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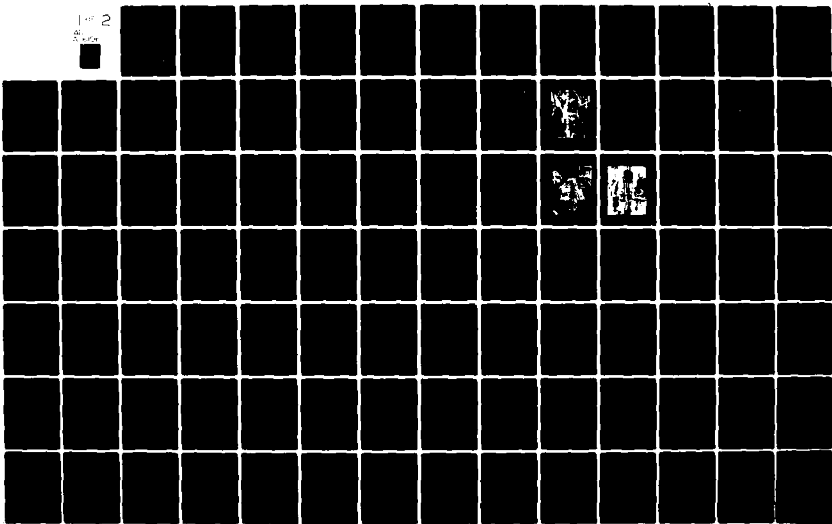

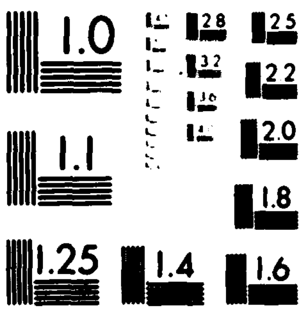

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ASD-TR-81-5033

INVESTIGATION OF "HYDRAULIC BUMP" IN SIMULATOR ELECTROHYDRAULIC CONTROLS

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AUGUST 1981

FINAL REPORT

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED

DEPUTY FOR ENGINEERING
AERONAUTICAL SYSTEMS DIVISION
AIR FORCE SYSTEMS COMMAND
WRIGHT-PATTERSON AFB, OHIO 45433

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This technical report has been reviewed and is approved for publication.

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ASD-TR-81-5033

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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER ASD-TR-81-5013	2. GOVT ACCESSION NO. AD-A112 106	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) INVESTIGATION OF HYDRAULIC BUMP IN SIMULATOR ELECTROHYDRAULIC CONTROLS		5. TYPE OF REPORT & PERIOD COVERED Final Report
		6. PERFORMING ORG. REPORT NUMBER FRC-F-C5306
7. AUTHOR(s) C.A. Belsterling, R.S. Fortner, J. Stone		8. CONTRACT OR GRANT NUMBER(s) F33657-80-C-0206
9. PERFORMING ORGANIZATION NAME AND ADDRESS Franklin Research Center, 20th & Benjamin Franklin Parkway, Philadelphia, PA 19103		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS 64227F-2325-21
11. CONTROLLING OFFICE NAME AND ADDRESS Aeronautical Systems Division (ASD/ENETS), Air Force Systems Command, Wright-Patterson AFB, Ohio 45433		12. REPORT DATE August 1981
		13. NUMBER OF PAGES
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for Public Release; Distribution Unlimited

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

Same

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Simulator
Electrohydraulic
Turnaround Bump
Smoothness
Acceleration Noise

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The report covers a series of experimental tests to determine the effect of cylinder and servovalve design on the magnitude of acceleration noise and hydraulic bump in simulator electrohydraulic controls. It includes tests on systems comparing equal with unequal area cylinders and high gain with low gain servovalves. The results indicate significant benefits with equal area cylinders and low gain servovalves.

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INVESTIGATION OF "HYDRAULIC BUMP" IN SIMULATOR ELECTROHYDRAULIC CONTROLS

INTRODUCTION

In the early 1970's the Franklin Research Center (FRC) initiated the development of the first electrohydraulic simulator [illegible] system. Over the next 10 years we have developed system components and design techniques that result in exceptional system performance. One outstanding characteristic is the smoothness of these systems, a result of the use of specially-designed servovalves and hydraulic cylinders. The qualitative benefits of these designs have been established through experience, but the magnitudes had not yet been quantified. The work described in this report was planned and executed to [illegible] and test typical motion system hardware to determine how much better the performance can be, compared with systems using standard servovalves. The work is a continuation of a previous project* in which the potential benefits were established with a small scale experimental system.

Testing in the previous project was done with negligible load mass on the actuator. This was done to emphasize the magnitudes of unwanted accelerations ("hydraulic bump") for series components of system performance. Part I of this report covers an extension of testing on the small scale system with a significant load mass added, yielding acceleration values typical of a small "[illegible]" system.

Part II of this report covers the development and test of a full scale single actuator system typical of present technology in simulator motion system design. Direct comparisons are made with the same system modified with a FRC [illegible] servovalve (design details are contained in Appendix A of Report [illegible]*).

Selected test results are contained in the body of the report to illustrate typical cases and to justify conclusions. The complete sets of test results are included in Appendices.

*Franklin Research Center Report No. [illegible] "Analytical and Experimental Studies on the Hydraulic Bump Phenomenon" Final Report on Contract No. [illegible], Sept. 1979.

Tests comparing the performance of a standard cylinder with a hydrostatic rod bearing against an FRC designed equal area cylinder will be published in a subsequent report.

CRITICAL DESIGN PARAMETERS

Based on past successful experience, the FRC staff has defined four design factors that must be satisfied to produce the smoothest high-performance electrohydraulic systems. They are each discussed in the following paragraphs.

Low Gain Servovalve

The hypothesis that a low sensitivity servovalve will reduce hydraulic bump is based on the fact that most high performance servovalves are designed with too much sensitivity in the final stage spool. They have evolved from World War II designs when electronic gain was costly in terms of size and power consumption (vacuum tubes). Therefore system designers produced servovalves with extremely high gains. One result was that the final stage spool became an item of high precision machining to match up spool lands with the outlet ports. The other result was that a very small displacement of the spool would open up considerable outlet area.

The design parameters critical to smooth, low velocity control are related to the final stage spool sensitivity. Conventional servovalves have ports in the body that are very narrow slots running circumferentially, that are intended to be precisely aligned with the edges of the lands on the movable spool. This arrangement has been chosen to provide maximum sensitivity (largest area versus spool displacement) in the final stage. This results in loss of control at low velocities due to the following. At low velocity the flow through the output ports is very low. As the spool moves through the neutral position to reverse actuator motion, an extremely small increment of spool displacement (< 0.001 inches) suddenly opens up a relatively large outlet area. Since there is little flow to cause a pressure drop, the full supply pressure appears at the output port and is transmitted through the piping to the actuator. This causes a sudden acceleration pulse (hydraulic bump) before the control system can react to avoid it.

The servovalve design that corrects for this unwanted condition involves a different design of the outlet ports in the final stage body. Instead of narrow circumferential slots we use narrow slots running longitudinally in the body wall. One end of each slot is aligned with the edges of the lands on the spool. In this arrangement, a relatively large spool displacement is required to open a considerable outlet area and the area is opened gradually in a smooth, controlled manner. Therefore the supply pressure is gradually metered to the reversing actuator and the control system can react normally. Because the final stage sensitivity (outlet area versus spool displacement) of this design is far less than that of the conventional design we call it the "FMC low gain servovalve."

For the purpose of comparing the effects of servovalve designs in the tests described in this report, FMC designed and fabricated a conventional and a low gain final stage valve, rated for the same flow in gallons per minute (gpm). The same two stage pilot valve was used to drive each spool as it was tested in the identical test rig, thereby isolating the effect of the different servovalve final-stage spool designs.

Equal Area Hydraulic Cylinder

The hypothesis that an equal area cylinder will reduce hydraulic bump is based on the fact that ordinary cylinders are designed with the rod connected to only one side of the piston. Because of this, the effective area on the rod side is considerably smaller than the effective area on the opposite side. Therefore when the servovalve spool goes through a null, there is a sudden discontinuity in the pressure-force relationship which produces an unwanted acceleration of the cylinder - hydraulic bump.

The FMC equal area cylinder design is illustrated in Figure 1.* The fixed portion on the left is made up of two concentric hollow tubes slightly shorter than the retracted length. The moving portion is a closed-end hollow tube with an annular flange that telescopes inside and between the fixed tubes. A seal is required around the moving tube and piston rings are required around its annular flange.

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*Figures begin on page 11.

The left cylinder port allows the hydraulic fluid to enter the fixed center tube and apply pressure on the closed end of the moving tube. The right cylinder port conducts fluid into the annular area between the moving tube and the fixed outer tube, and applies pressure on the annular area projected by the flange. By proper selection of design dimensions, it is clear that the effective area of the closed end of the moving tube can be made equal to or different from the effective area of the annular flange.

Preloaded Cylinder Connections

In some electrohydraulic motion systems, conditions can be encountered when the reaction force on the electrohydraulic actuator goes through zero. If there is lost motion or backlash in the mechanical connections between the cylinder and the load, the actuator rod will accelerate and impact against the limit of lost motion. This obviously will create the effect of a hydraulic bump whether or not the control system can react to it. Therefore a design for a smooth motion system must provide for preloading of the cylinder connections or through the use of preloaded connection hardware.

For the purposes of the tests described in Part II of this report, the cylinder connections are continually preloaded by static forces.

Low-Friction Cylinder Design

Finally, the most commonly misunderstood cause of unwanted hydraulic bump is sticking friction. As the actuator is reversing direction it comes to a complete stop; while the hydraulic force continues to rise, it won't move until the friction force is overcome. When it is, the cylinder rod accelerates excessively, resulting in an effective bump in the motion.

To avoid excessive sticking friction, FMC fits all cylinders with low friction piston rings and rod seals. In our experience this is sufficient to keep the unwanted accelerations to an exceptionally low level. Other designers, however, believe that hydrostatic bearings are necessary on the rod seals and on the piston rings. The [illegible] Advanced Simulator Technology [illegible] cylinder used in the full scale tests described in this report included a hydrostatic rod bearing.

PART I. SMALL SCALE SYSTEM TESTS

Description of the Small Scale Test System

Figure 2 is a photograph of the small scale test system with a 350 lb. load mass. With reference to Figures 3 and 4 we will describe the hydraulic system used in the small scale system tests. Two standard hydraulic cylinders with a bore of 2 1/2 inches and a stroke of 8 inches are connected opposing in a horizontal position on a heavy bed plate. A supply of hydraulic fluid at 2000 psi is fed to the servovalve under test. The return from the servovalve flows through a calibrated check valve to maintain a return pressure of 50 psi.

For unequal cylinder area tests, the cylinder ports are connected to the servovalve as shown in Figure 3 providing a ratio of about 3:1. One small end cylinder port is connected to an accumulator which can be charged to provide a preloading force on the cylinder connections.

For the equal cylinder area tests the large end cylinder ports are connected as shown in Figure 4, preloading the cylinder mechanical connections under normal conditions. Accumulators are connected to the small end ports to provide a force to unload the mechanical connections when desired.

Pressure transducers are provided on each of the servovalve outlet ports. A force transducer is inserted into the pin coupling the two opposing cylinder rods. A Linear Variable Differential Transformer (LVDT) is attached to the servovalve spool to measure displacement. An acoustic feedback transducer is attached to the cylinder rod to measure cylinder displacement. An accelerometer is mounted on the mechanical connection between the opposing cylinder rods to measure total accelerations.

The small scale test system was tested in four combinations of servovalves and cylinder areas. The following tests were run on each.

Waveform	Frequency
Stationary	0.00 Hz
Sinusoidal	0.01 Hz
Sinusoidal	0.05 Hz
Sinusoidal	0.10 Hz

Waveform	Frequency
Sinusoidal	0.20 Hz
Sinusoidal	0.50 Hz
Sinusoidal	1.00 Hz
Triangular	0.01 Hz
Triangular	0.10 Hz
Triangular	0.20 Hz
Square Wave	0.20 Hz

Small Scale System Test Results

The results of previous tests with negligible load mass on the actuator are illustrated in Figures 5 and 6, repeated here for comparison with new tests. Note that there is a significant difference in the magnitudes of unwanted accelerations, the commercial valve and unequal cylinder configuration having peaks of 0.25g (Figure 5).

Figures 7 through 14 are the results of tests on the same small scale system but with a 250 lb. load added. One second timing marks are on the bottom of the charts.

Comparisons with Figures 5 and 6 show 1) the magnitudes of the unwanted accelerations produced by unwanted hydraulic forces are substantially reduced by the load mass (acceleration = $\frac{\text{force}}{\text{mass}}$) and 2) there is still a significant difference in magnitudes, the commercial valve with unequal cylinder areas reaching peaks of 0.05g at constant velocity (Figure 9) as compared with peaks of 0.005g with the Franklin valve (Figure 10).

Figures 13 and 14 illustrate the effect of unloaded cylinder end couplings. Both competitive systems exhibit large "hydraulic bumps".

Figures 7 through 14 are representative of the tests run on the small scale test system with load. The complete set of tests is contained in the Appendices identified as follows:

Appendix A. Small Scale System Tests
Commercial High Gain Valve with Unequal Cylinder Areas

Appendix B. Small Scale System Tests
Franklin Low Gain Valve with Unequal Cylinder Areas

Appendix C. Small Scale System Tests
Commercial High Gain Valve with Equal Cylinder Areas

Appendix D. Small Scale System Tests
Franklin Low Gain Valve with Equal Cylinder Areas

Analysis of the Small Scale System Test Results

Figures 5 and 6 are the results of previous tests on the small scale system without a significant load mass. This arrangement was intended to emphasize the accelerations due to unwanted forces developed in the hydraulic cylinder.

Figures 5 and 6 show a direct comparison of small scale systems performance at 0.20 hz. In this case the unwanted accelerations at turnaround are 0.35g for the high gain valve and 0.10g for the low gain valve. The high gain valve produces an unwanted acceleration more than three times greater than the low gain valve.

The next series of tests were done on the same small scale system, but with a 150 lb. load mass added. The mass was supported on four linear ball bearings riding on ground guide rods. This eliminated any side loading on the cylinders and minimized friction. The accelerometer for measuring the resulting accelerations was mounted on the load mass. This arrangement effectively acts to attenuate unwanted accelerations due to spurious forces developed in the cylinder.

Figures 7 and 8 compare the behavior of the two competitive servovalves in the loaded small scale system at a frequency of 0.10 hz. Note that the low gain valve provides results indicating a lower level of turnaround hump (0.02g vs. 0.03g). Comparison of Figures 7 and 8 with Figures 5 and 6 show the effect of the addition of the load mass. The accelerations are attenuated by a factor of more than 10.

Figures 9 and 10 show the performance of the two servovalves at a very low constant velocity (0.10 in/sec), when an unwanted acceleration would be most noticeable to a human subject. In this case the low gain valve keeps

the unwanted accelerations to less than 0.005g (Figure 10) as compared with nearly 0.05g with the high gain valve.

Figures 11 and 12 are the results of competitive tests at a frequency of 0.50 hz. Again the comparison shows that the peak accelerations at turnaround are approximately half as great when using the low gain servovalve.

Figures 13 and 14 illustrate the effect of unloaded cylinder couplings. Note that peak accelerations of nearly 0.30g are generated regardless of the characteristics of the servovalves.

Summary and Conclusions

The small scale electrohydraulic system was assembled under a previous Air Force project to demonstrate the performance of two competitive servovalve designs. That system was designed with a minimum of moving mass to emphasize the unwanted accelerations of the actuator during operation. The test results indicated that the Franklin low gain servovalve delivered system performance with significantly lower unwanted accelerations than the commercial high gain servovalve.

In Part I of the project described in this report, a load mass of 150 lb. was added to the original small scale test system. The tests run under the previous Air Force project were repeated and similar results recorded. These results confirm that the system containing the low gain servovalve delivers superior performance compared with the system containing the high gain servovalve. They also illustrate the need for preloading cylinder end connections. Comparison of these results with the results of the previous tests show that the additional load mass on the system acts to attenuate unwanted accelerations by a factor of nearly 10.

PART II. FULL SCALE SYSTEM TESTS

Description of the Full Scale Test System

The circuit schematic of the full scale test rig is shown in Figure 15. The hydraulic supply of 1200 psi is fed to the servovalve with an accumulator to smooth pump pulsations. The return from the servovalve flows through a calibrated check valve to maintain a 50 psi return pressure. The servovalve outlet ports are connected through equal lengths of piping to the appropriate cylinder ports. The hydrostatic bearing is fed from the same supply through separate lines.

Pressure transducers are mounted at the outlet ports of the servovalves. An LVDT is mounted on each servovalve spool to measure displacement. An acoustic feedback transducer is mounted inside the rod of the cylinder to measure actuator displacement. An accelerometer is mounted on the arm supporting a 4000 lb. load to measure simulated accelerations at the pilot's station. It also detects components of gravity as the support arm moves through an arc.

The second generation test rig is designed to represent, as nearly as practical, a typical simulator electrohydraulic actuator with realistic loading. It includes a Link Advanced Simulator Technology cylinder with a hydrostatic end bearing loaned to FRC specifically for these tests. It is designed to operate on 1200 psi with a bore of 4 inches, a stroke of 36 inches and a length at neutral of 128 inches.

The full scale test rig is shown in Figure 16. The cylinder is nominally horizontal, acting through a crank mechanism to support a total mass of approximately 4150 lbs. The mass is reflected as a static load on the cylinder as if it were vertical and supporting its share of a typical load on a synergistic six-degree-of-freedom motion system. We believe that the effects of a difference in side loading on the cylinder due to the different vector components of gravity is not significant in our comparative tests.

The Link actuator has a valve manifold located adjacent to the clevis-end port, connected through a length of hard piping to the rod end port.

This arrangement can lead to unwanted accelerations because of the non-symmetry of the hydraulic circuit. Therefore a new valve manifold was fabricated and mounted midway between the cylinder ports as shown in Figure 17.

The full scale test rig was evaluated in two configurations: 1) with the commercial high gain servovalve and 2) with the FMC low gain servovalve, both units built during the previous project. The following set of tests were run on each:

Waveform	Frequency
Stationary	0.000 Hz
Sinusoidal	0.01 Hz
Sinusoidal	0.05 Hz
Sinusoidal	0.10 Hz
Sinusoidal	0.20 Hz
Sinusoidal	0.50 Hz
Sinusoidal	1.000 Hz
Triangular	0.01 Hz
Triangular	0.10 Hz
Triangular	0.20 Hz
Square Wave	0.20 Hz

Stabilization of the Test System

After fabrication and assembly of the full scale test system, a number of tests were performed to isolate the resonant elements, then stiffen the structure and dampen the hydromechanical dynamic response. The system could then be properly stabilized, but there remains a lightly damped mechanical resonance in the support structure at 20 Hz that amplifies the accelerations measured at the accelerometer station.

Full-Scale System Test Results

Figures 18 and 19 show the square wave response of both competitive systems. This proves that the system minor loop and major loop gains are

nearly the same, thereby isolating the differences in performance to the differences in design of the servovalve main spool. One second timing marks appear at the bottom of the charts.

Figures 20 and 21 show the response of both competitive systems in sinusoidal motion at 0.05 hertz. This is by far the most critical case because the pilot is most sensitive to unwanted accelerations when the wanted accelerations are near his threshold of perception.

Figures 22 and 23 show the response of both competitive systems during constant velocity (triangular wave motion) at 0.20 in/sec. This case simply illustrates the smoothness of the components of the system other than the servovalve which is biased open in one direction or other. It also allows the mechanical vibrations to decay and reveal the true smoothness of operation.

Figure 24 and 25 show the response of both competitive systems in sinusoidal motion at 0.5 hertz. This is the frequency that is used in the smoothness tests required under Mil Standard 1558. This commanded frequency is mainly exciting the mechanical resonance at 20 hz.

Finally we show Figures 26 and 27 recording the response of both competitive systems at standstill (zero steady command signal). They indicate the outstanding ability of both servovalves to respond and correct for random disturbances.

Figures 18 through 27 are samples of the complete set of tests required to compare the relative performance of the commercial high gain servovalve design with the FRC low gain servovalve design. The complete set is contained in the Appendices, identified as follows:

Appendix E.	Full Scale System Tests Commercial High Gain Valve
Appendix F.	Full Scale System Tests Franklin Low Gain Valve

Analysis of the Full Scale System Test Results

Figures 18 through 27 show the results of tests on the full scale system assembled to compare performance with the commercial high gain servovalve and the Franklin low-gain servovalve.

Figures 18 and 19 show the response of the system using competitive servovalves to a square wave command signal with a period of 3 seconds. It illustrates the relative dynamics of the two systems. It should first be noted that the response contains a predominant frequency around 20 hertz, which has been identified as the resonant frequency of the structure supporting the 5150 lb. load. Note also that it is very lightly damped. The accelerometer is mounted on this structure, therefore the measured response contains this resonant frequency which, when excited by commanded motions, can mask the true unwanted accelerations.

The responses to the square wave command are similar for the systems containing the competitive servovalves. This confirms that the minor and major feedback loops have been adjusted to compensate for their different gains.

Figures 20 and 21 are recordings at the lowest sinusoidal test frequency where an effective hydraulic bump is noticeable. This is a critical case when the simulator pilot is most perceptive to unwanted acceleration. At this frequency the peak of the unwanted acceleration in the system using the low gain valve is 0.0075g compared to 0.0125g for the high gain valve.

Figures 22 and 23 show the performance of the system using the competitive servovalves for the case of a very low (0.20 in/sec) constant velocity. Again this is a critical case when the pilot is most perceptive to unwanted accelerations. Comparison of these figures shows no significant difference between system performances. With both servovalves the unwanted peak accelerations at constant velocity are 0.0038g.

Figures 24 and 25 illustrate system responses at a sinusoidal frequency of 0.50 hertz. In this case the commanded motion has a peak acceleration of 0.015g which simply excites the resonant frequency of the

support structure. Therefore the unwanted accelerations cannot be separated from the total acceleration as recorded.

Figures [illegible] and [illegible] show the test results of the system containing competitive servovalves with a zero command signal. In this case the test results are a measure of the long term stability of the system. The results indicate that over a period of more than [illegible] minutes both systems were extremely quiet, with unwanted peak accelerations of not more than [illegible] g. There is no significant difference between the two systems.

Summary and Conclusions

The objective of this part of the project was to build and test a full scale electrohydraulic actuator system in a configuration representative of a realistic simulator motion system. This has been accomplished using a full size [illegible] cylinder mounted in a test rig with a [illegible] load mass alternately controlled by two servovalves of different design. The first is a commercial design with a high gain final stage spool. The second is a Franklin design with a low gain final stage spool. A full range of tests were performed to compare the performance of the servovalves, particularly their ability to minimize unwanted accelerations in a low acceleration environment.

The test results indicate that both servovalves provide outstanding stability at standstill ([illegible]g) and smoothness at constant velocity ([illegible]g). The results also indicate that, under the most critical conditions the Franklin low gain servovalve exhibits significantly less hydraulic turnaround bump than the commercial high gain design ([illegible] vs. [illegible]g). In light of the foregoing and the small scale system tests, which clearly show the advantages of the low gain servovalve, the use of the low gain design in flight simulator motion systems is recommended.

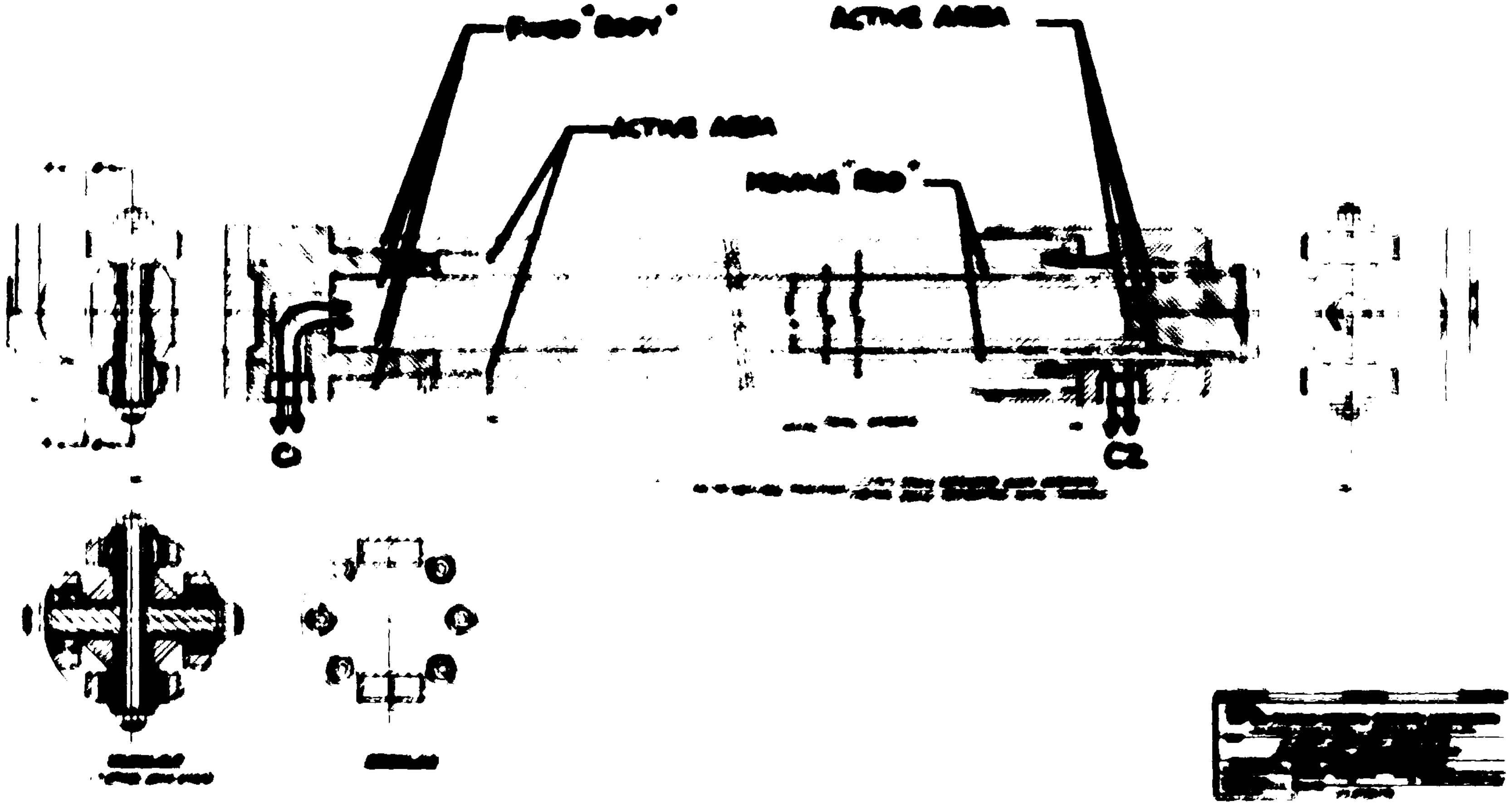


Figure 1. Typical Franklin Equal Area Cylinder

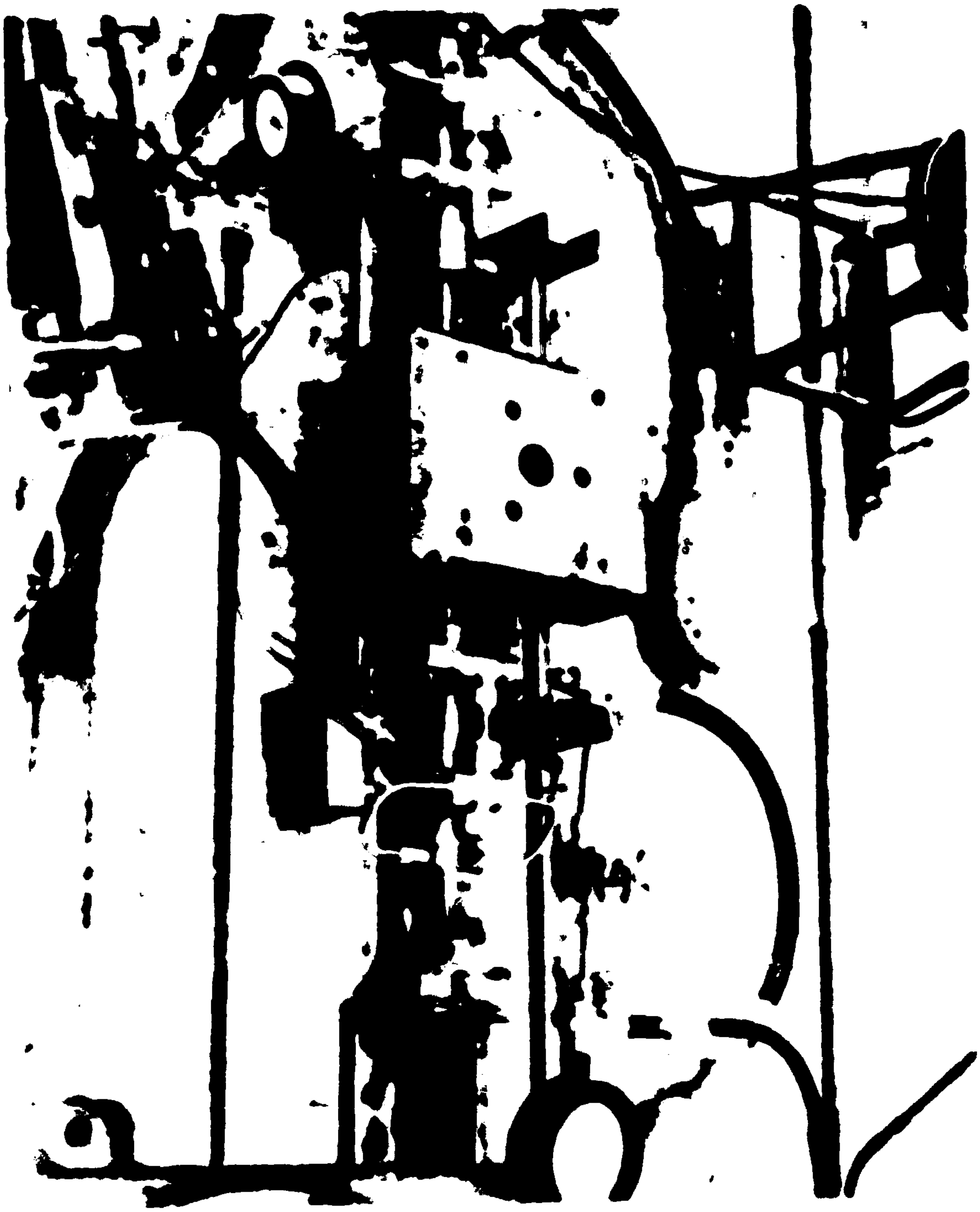


Figure 2. Photograph of Small Scale Test System

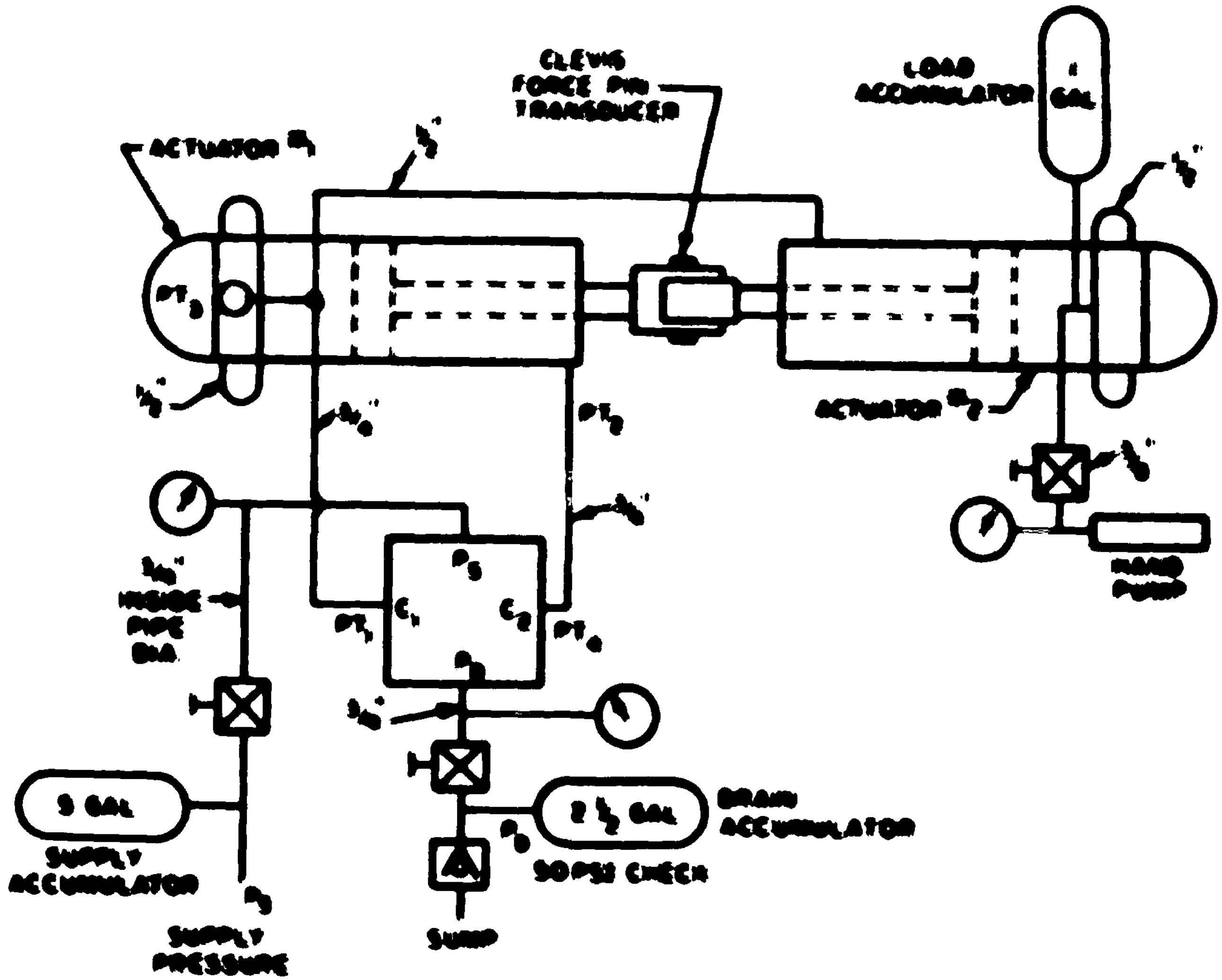


Figure 2. Small Scale Test Circuit, Unequal Cylinder Areas

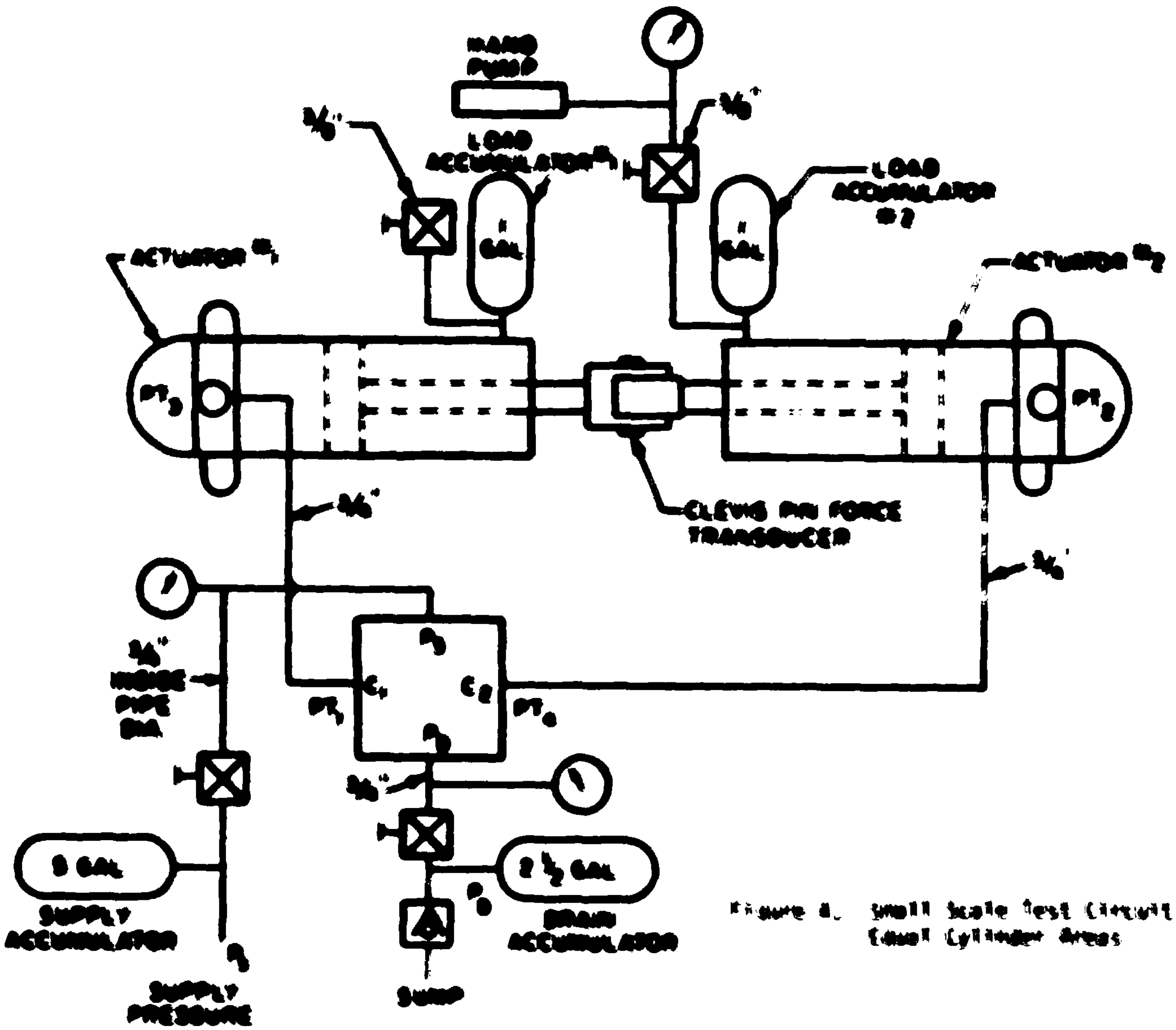


Figure 4. Small Scale Test Circuit, Equal Cylinder Areas

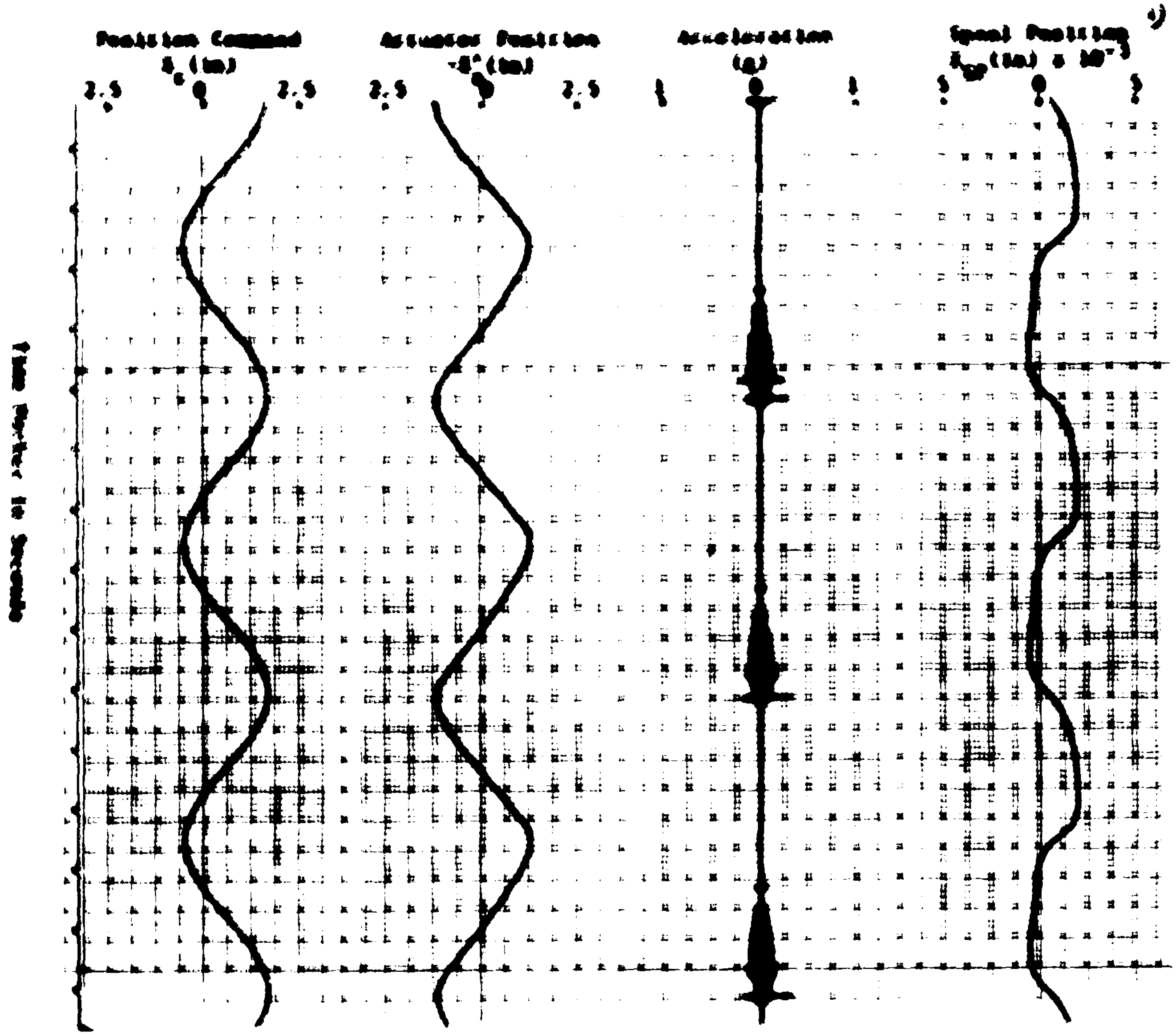


Figure 5. No Load Mass, High Gain Value, Unequal Cylinder Areas, 0.20 Hz

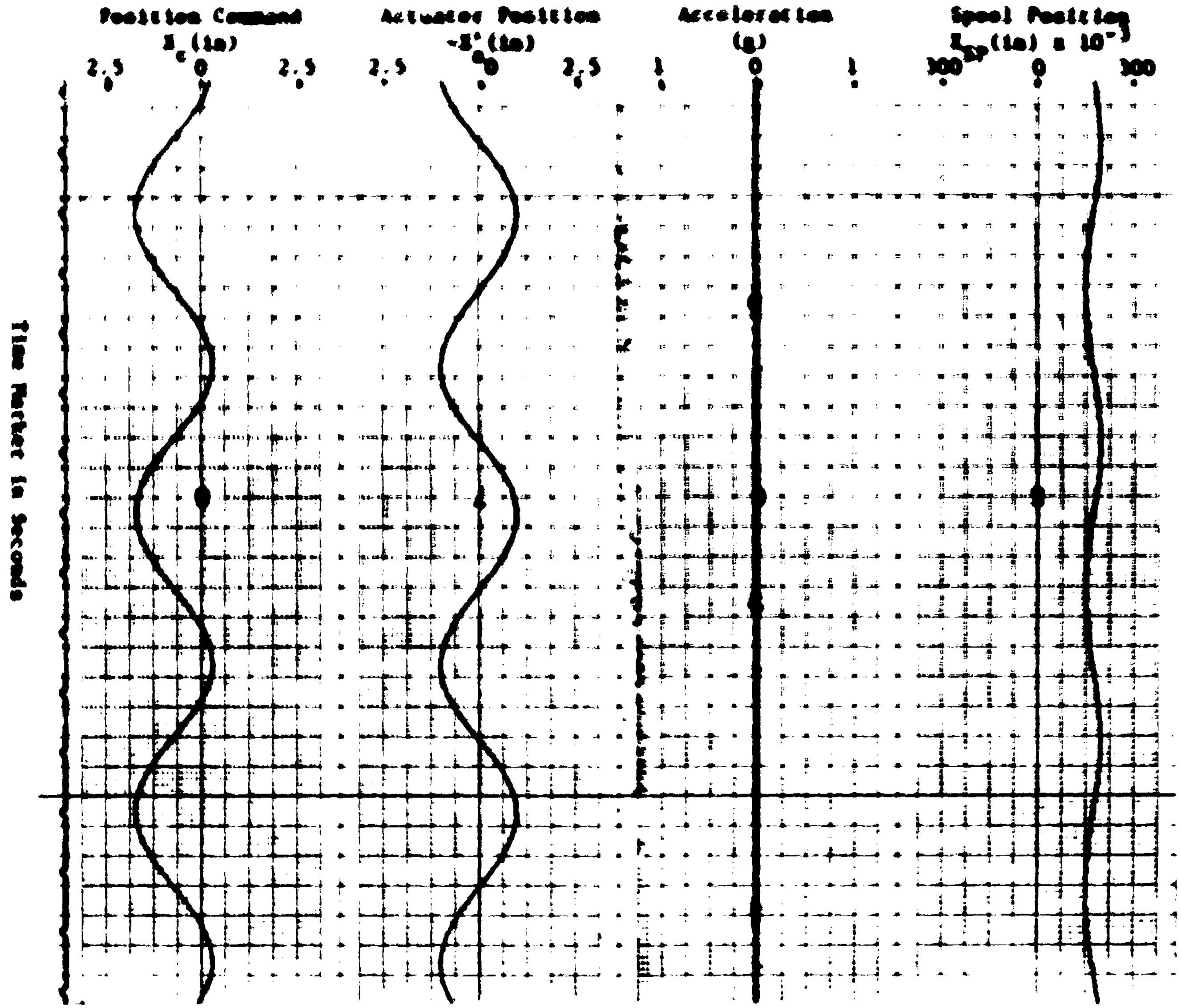


Figure 6. No Load Mass, Low Gain Valve, Unequal Area Cylinders, 0.20 Hz

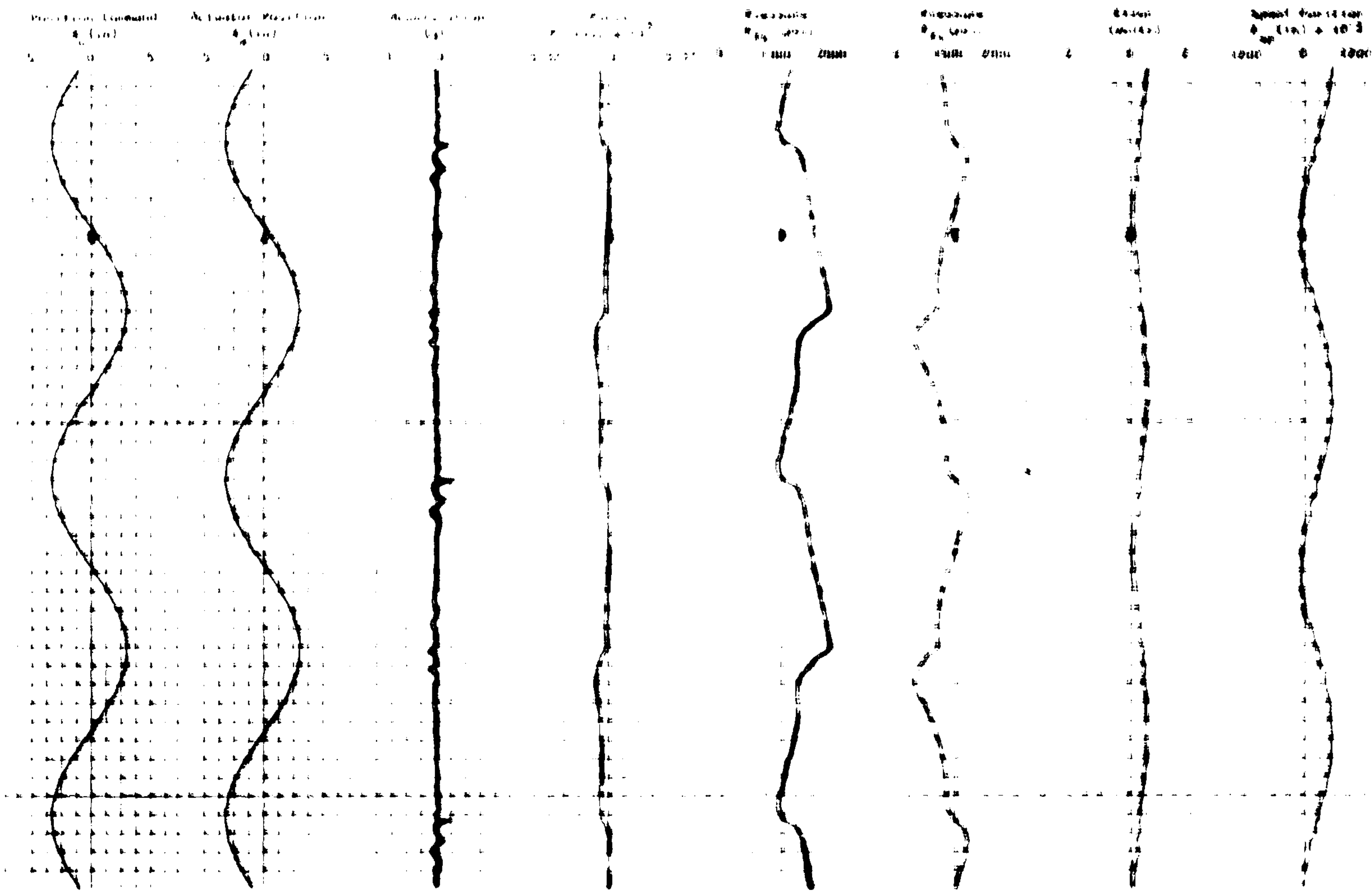


Figure 7. 350 Lb. Load Mass, High Gain Valve, Equal Cylinder Areas, 0.10 Hz

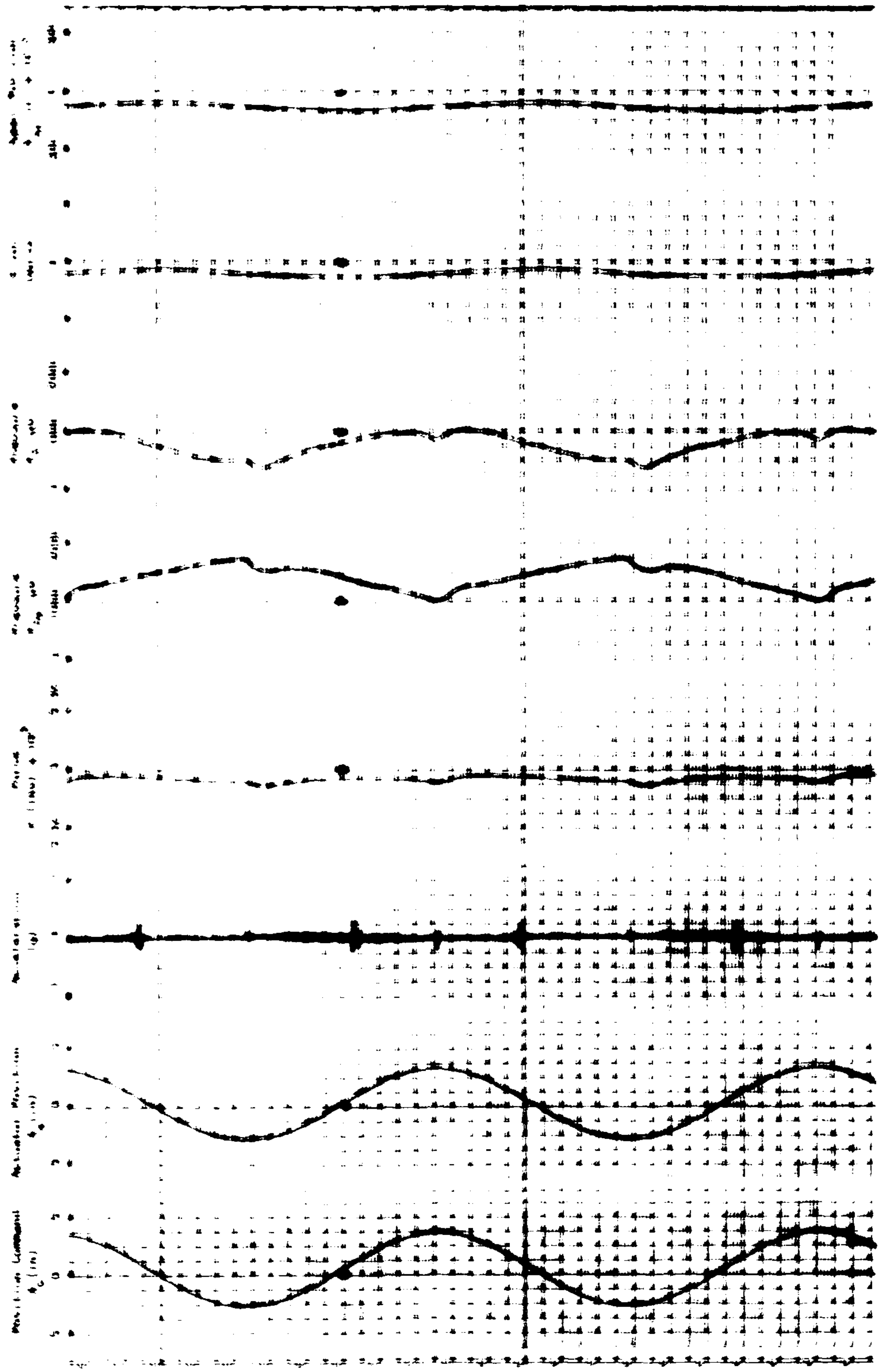
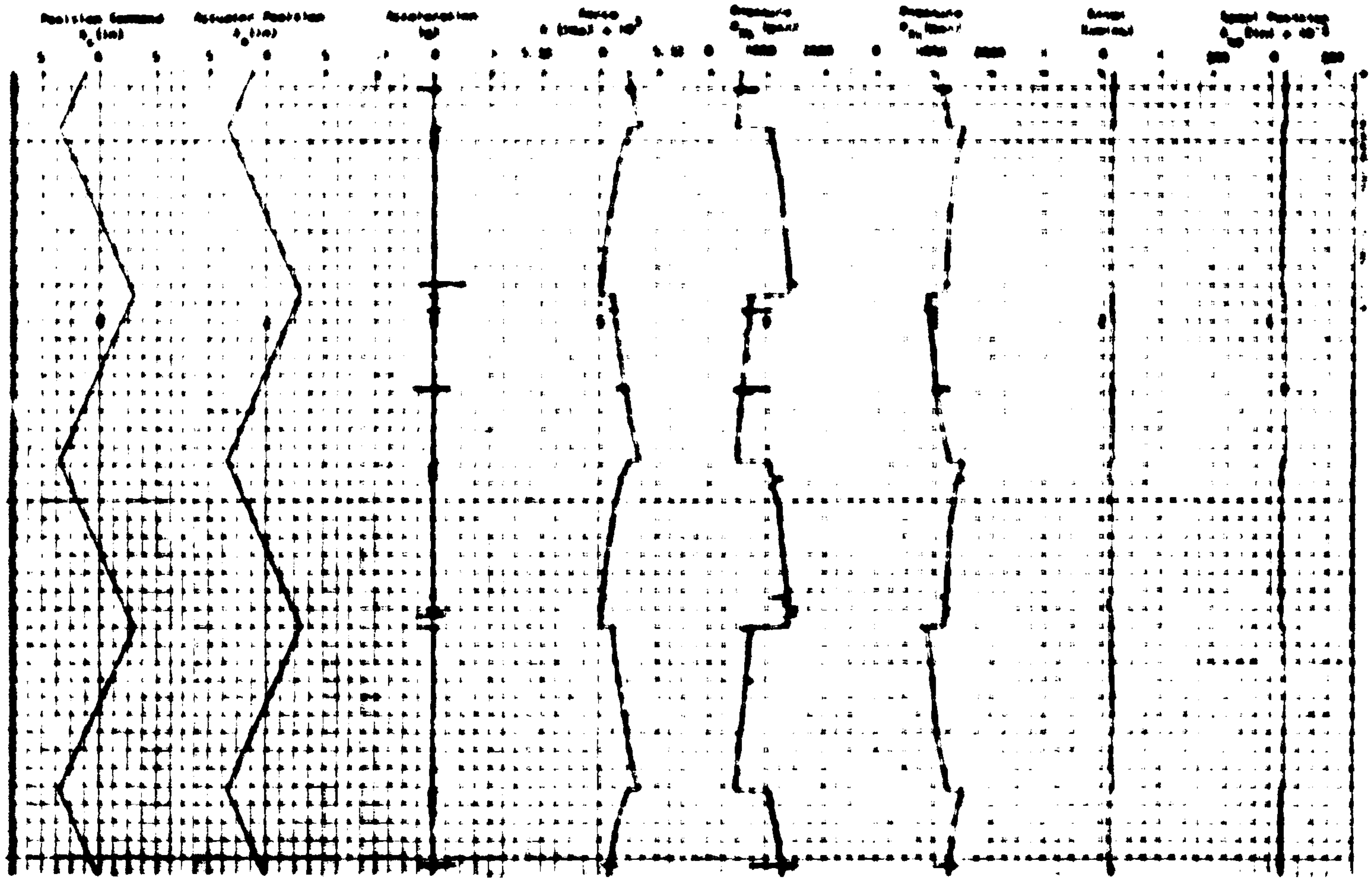


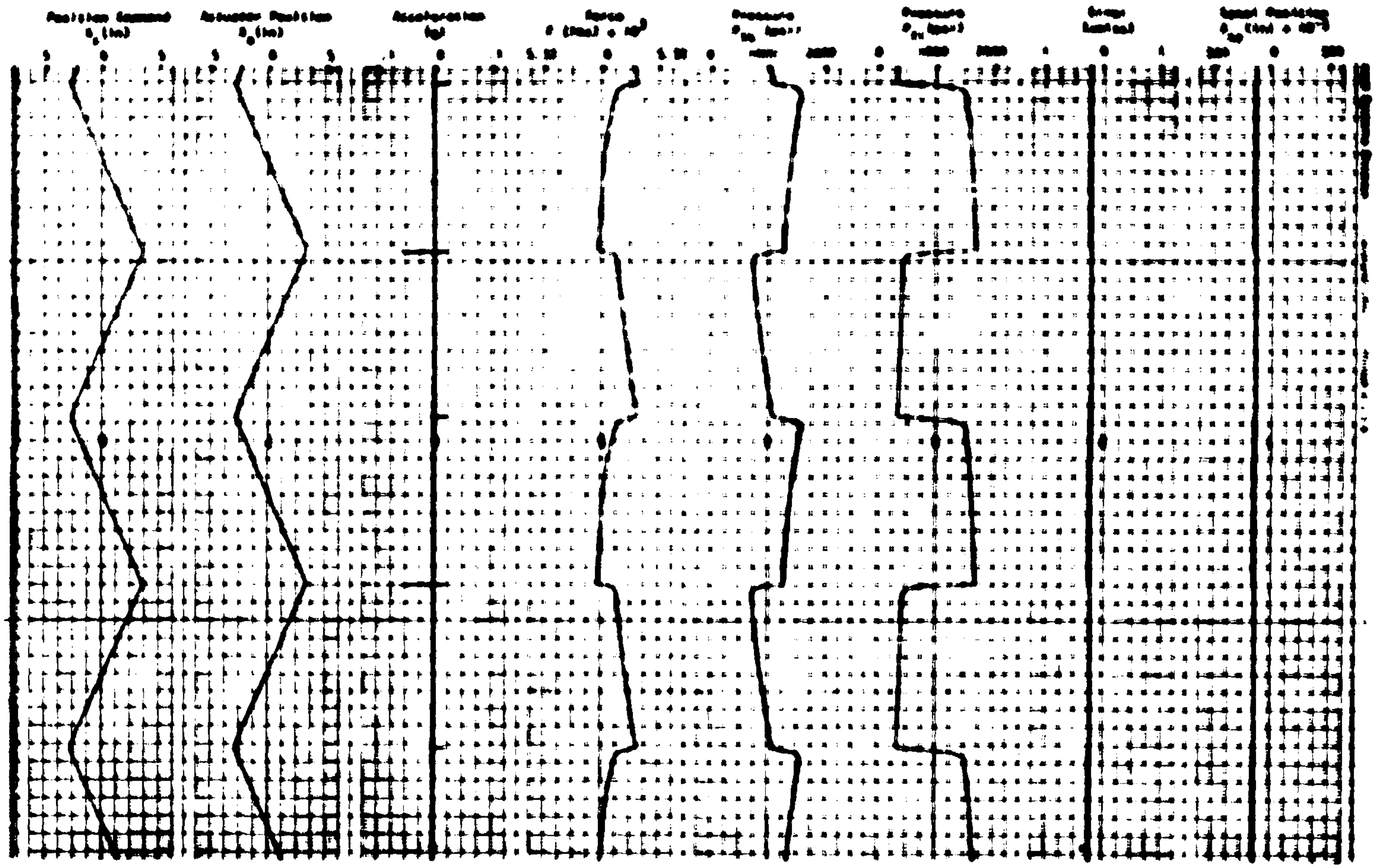
Figure 8. 350 Lb. Load Mass, Low Gain Valve, Equal Cylinder Areas, 0.10 Hz



FREQUENCY: 0.01 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

Figure 9. 350 Lb. Load Mass, High Gain Valve,
Unequal Cylinder Areas,
Constant Velocity = 0.10 in/sec.



FREQUENCY: 0.01 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

Figure 10. 350 Lb. Load Mass, Low Gain Valve,
Unequal Cylinder Areas,
Constant Velocity = 0.10 in/sec.

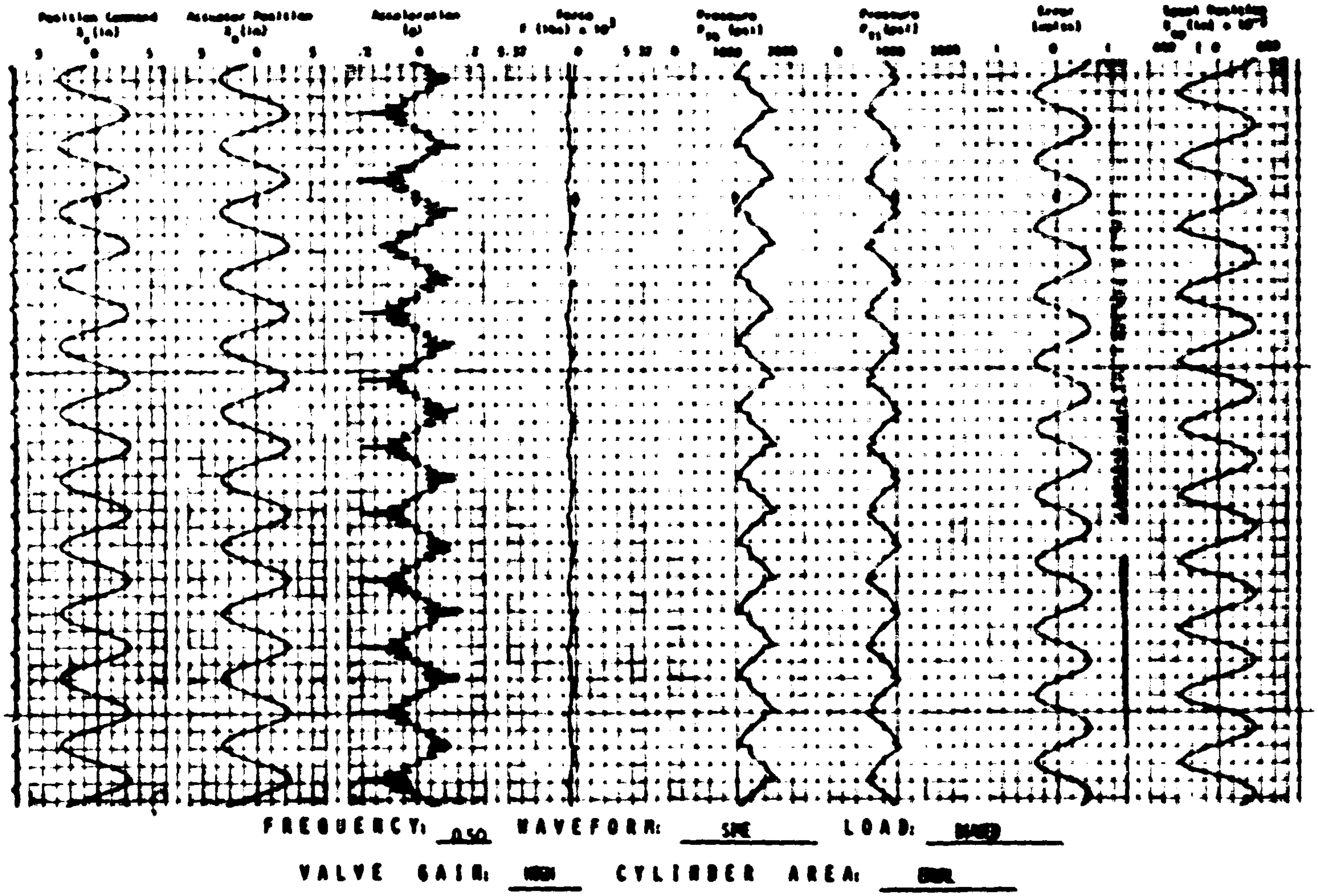
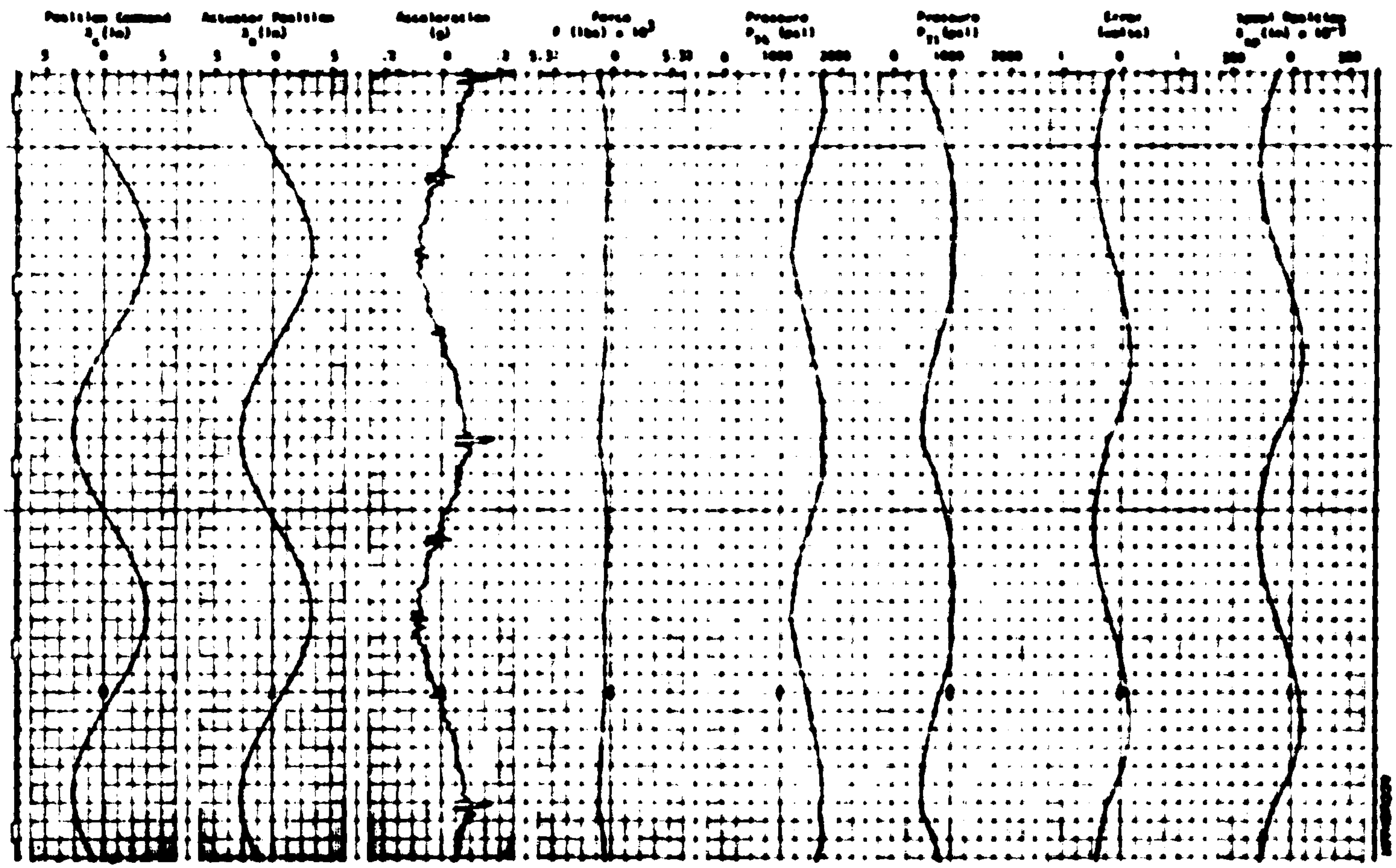


Figure 11. 350 Lb. Load Mass, High Gain Valve, Equal Cylinder Areas, 0.50 Hz



FREQUENCY: 0.50 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

Figure 12. 350 Lb. Load Mass, Low Gain Valve, Equal Cylinder Areas, 0.50 Hz

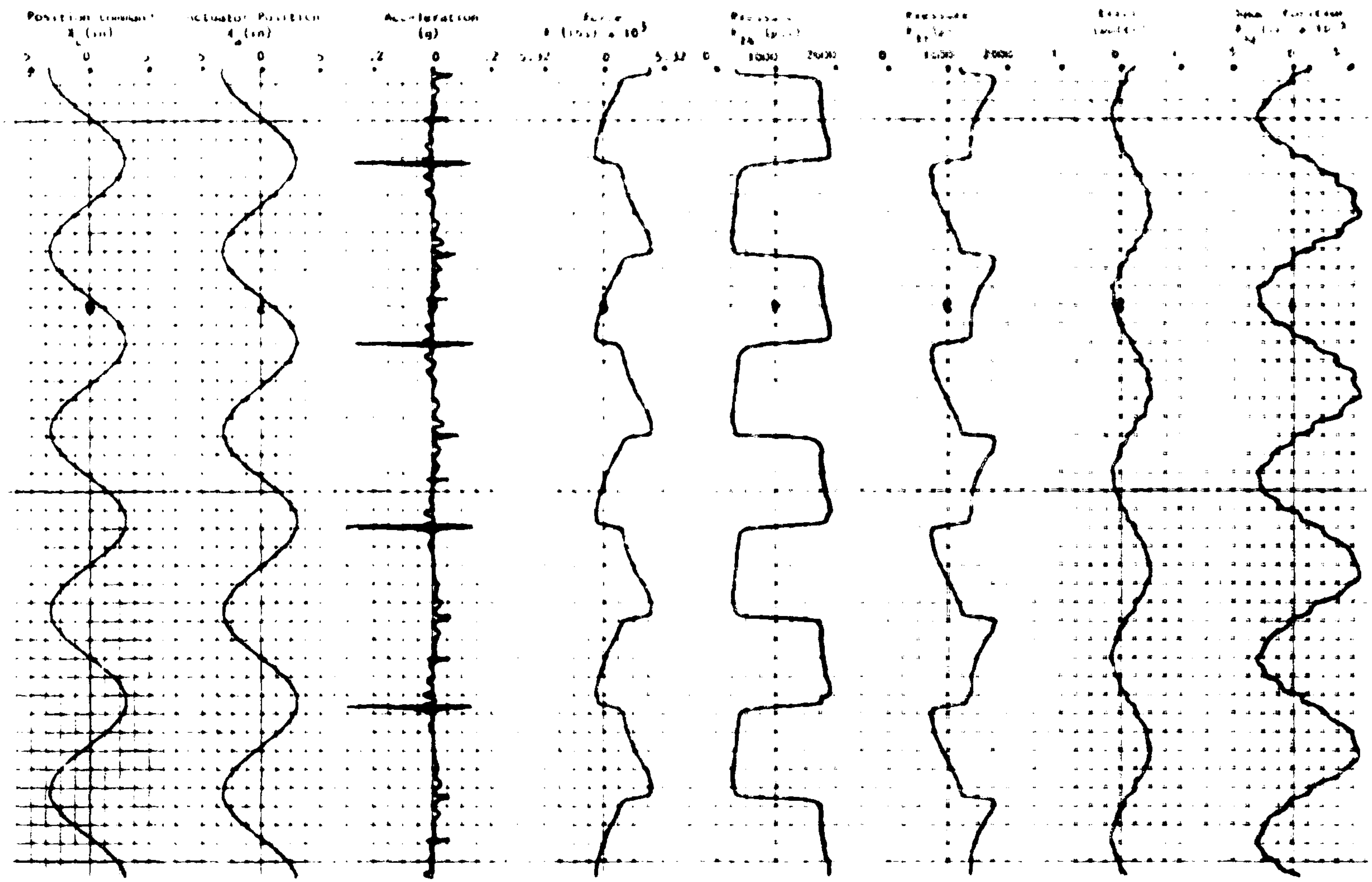


Figure 13. 350 Lb. Load Mass, High Gain Valve, Unequal Cylinder Areas, 0.20 Hz

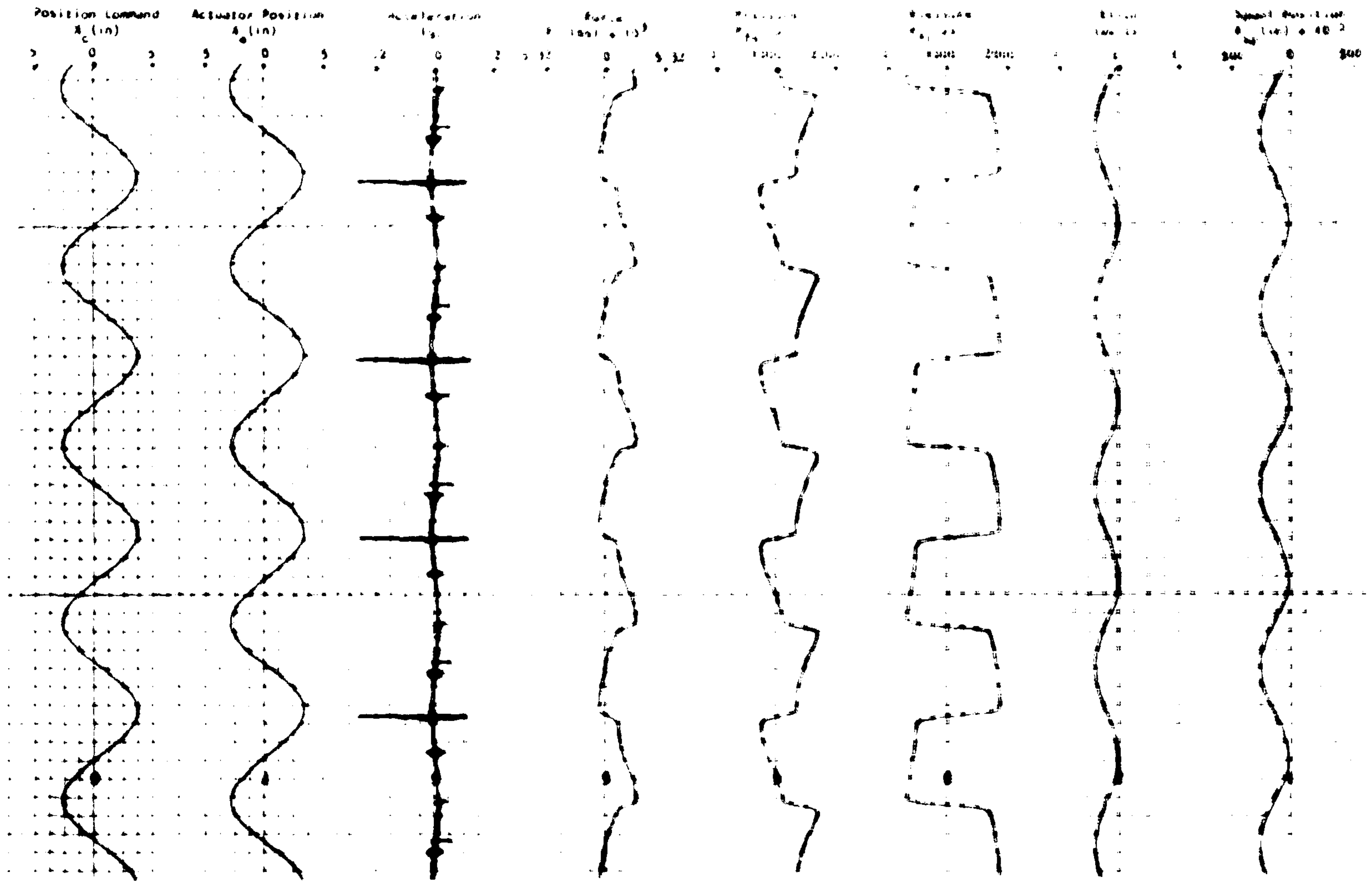


Figure 14. 350 Lb. Load Mass, Low Gain Valve,
Unequal Cylinder Areas, 0.20 Hz

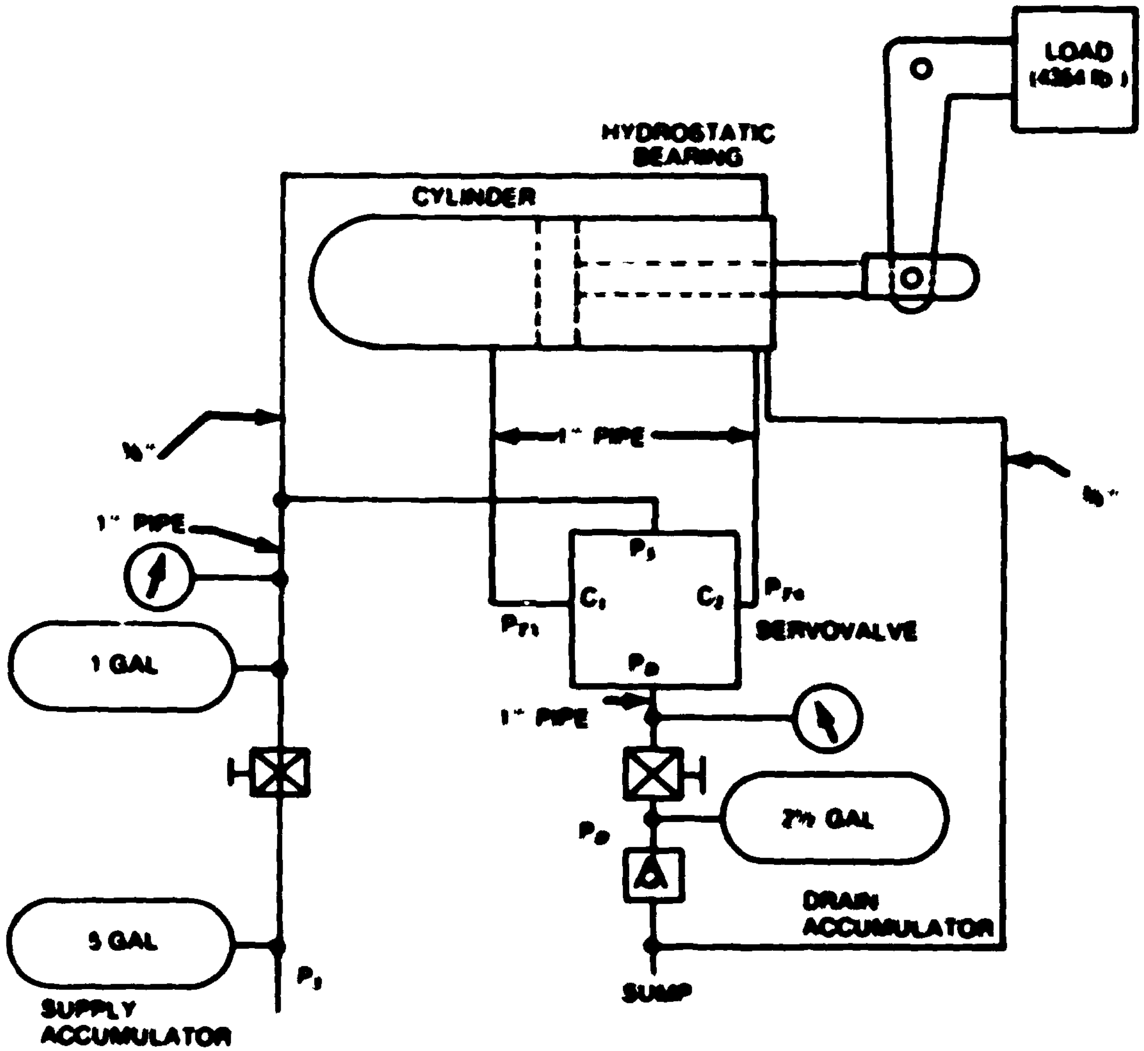


Figure 15. Full Scale Test Circuit

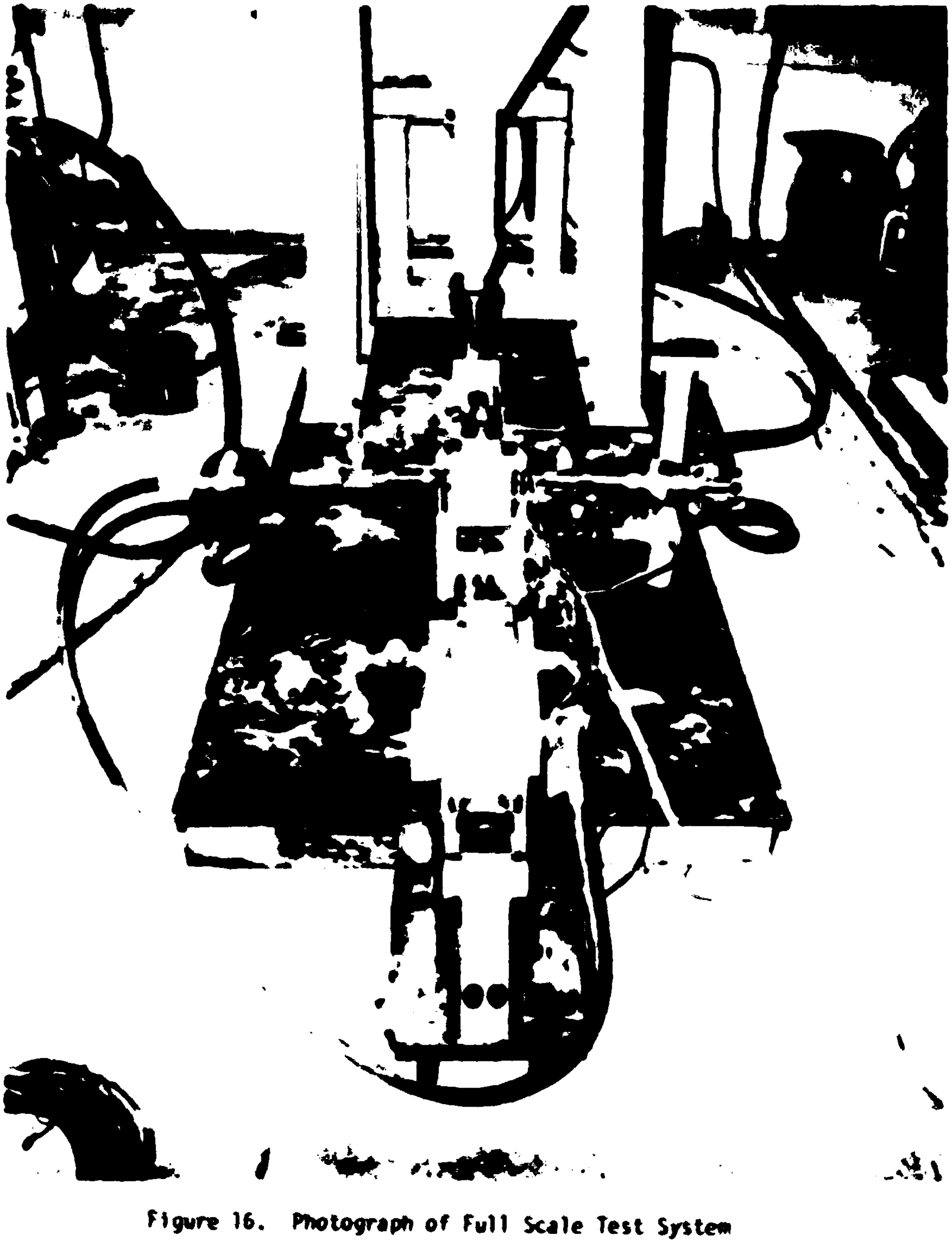


Figure 16. Photograph of Full Scale Test System

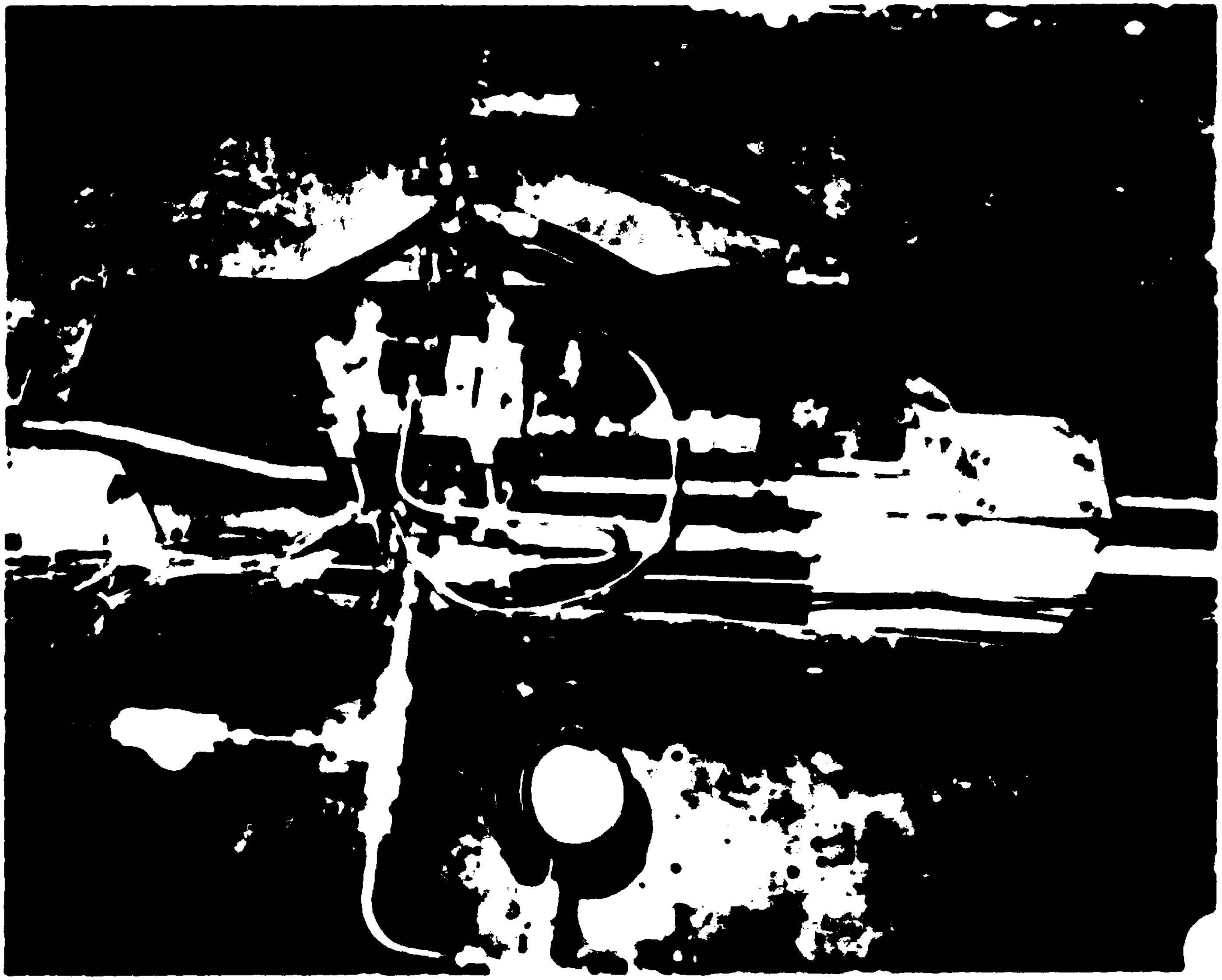


Figure 17. Photograph Showing Location of the Servovalve

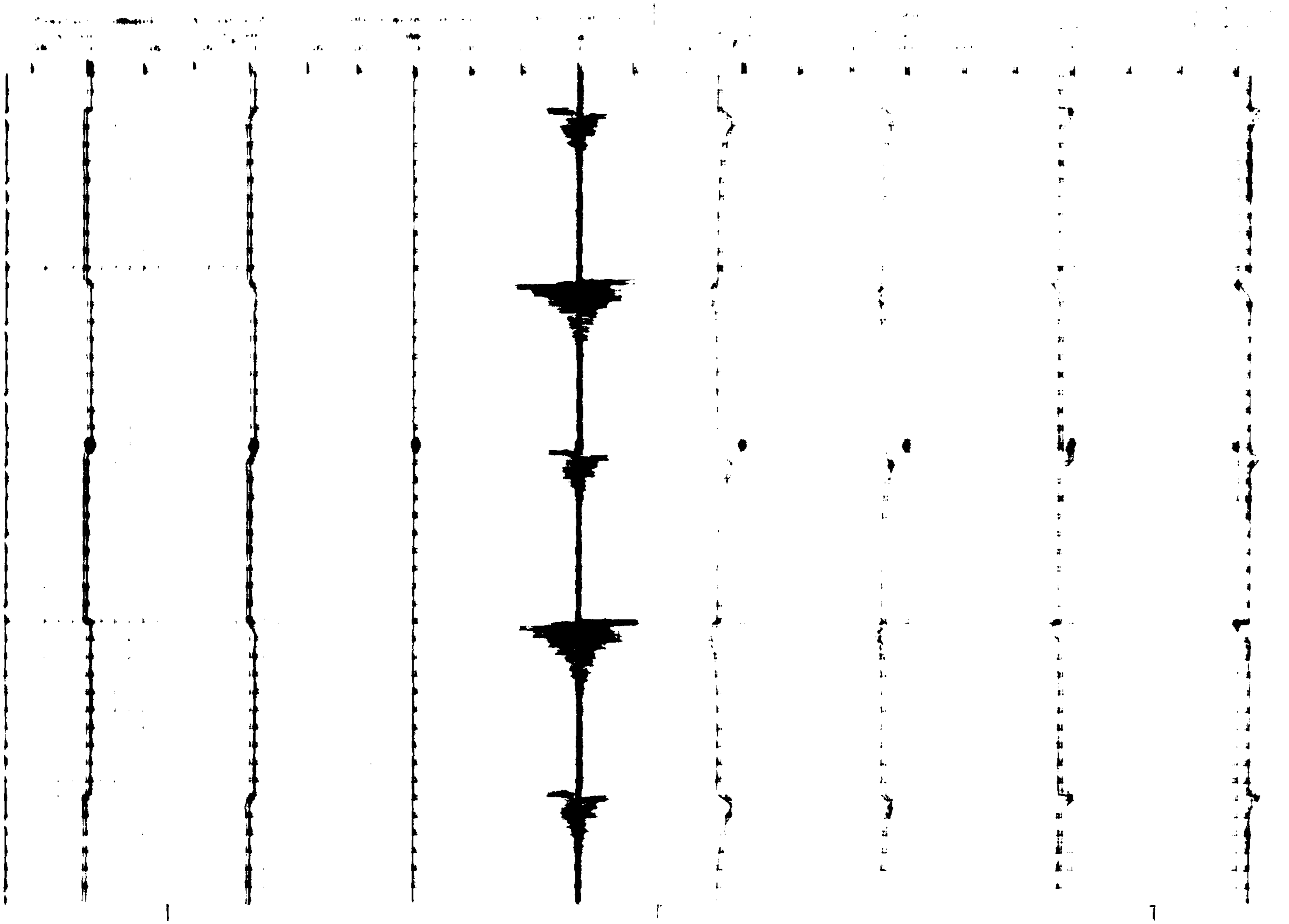


Figure 14. Full Scale Test, Commercial High Gate Valve, 1.20 Hz Square Wave

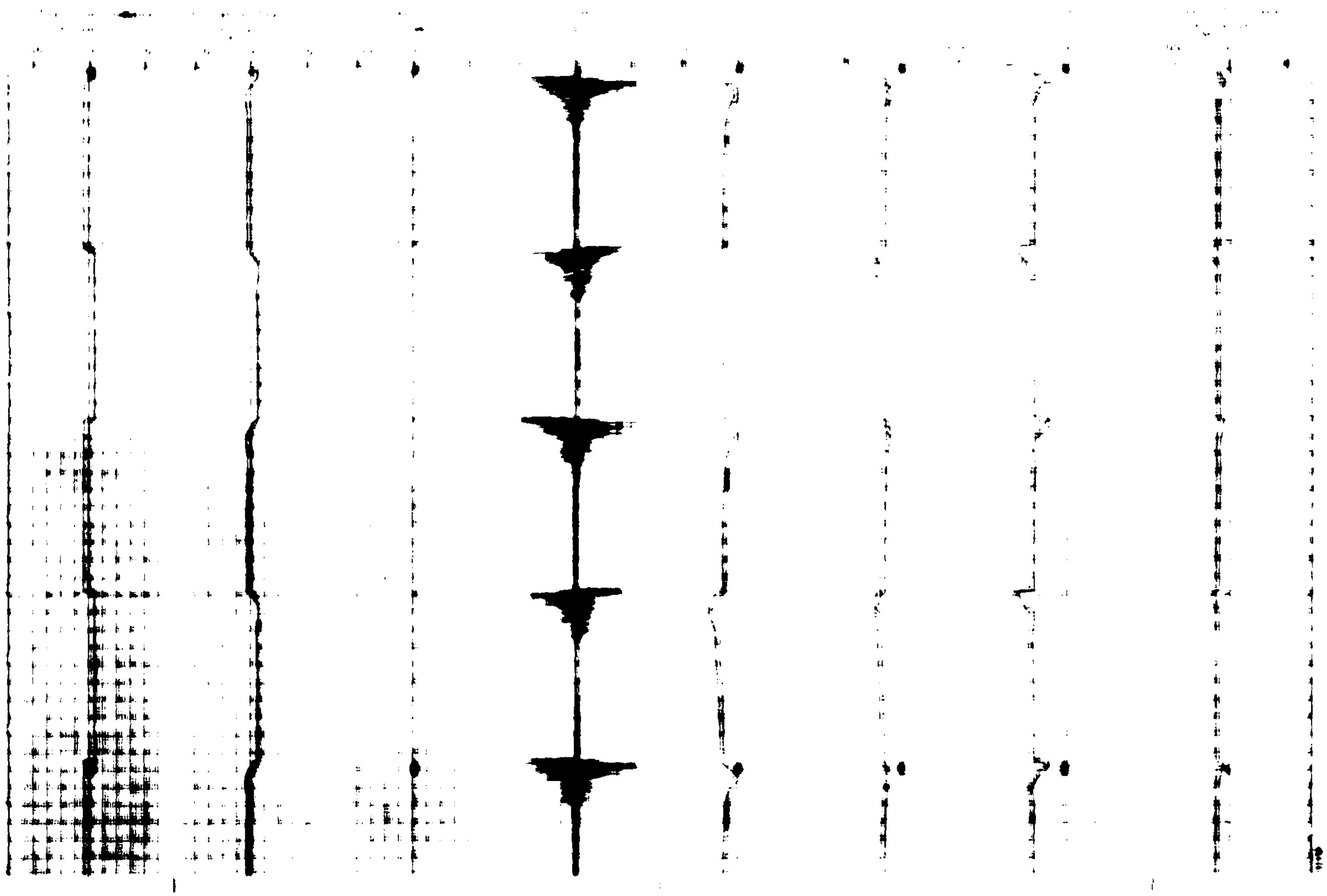


Figure 13. Full Scale Test, Repeatable Low Gain Valve, 0.20 Hz Square Wave

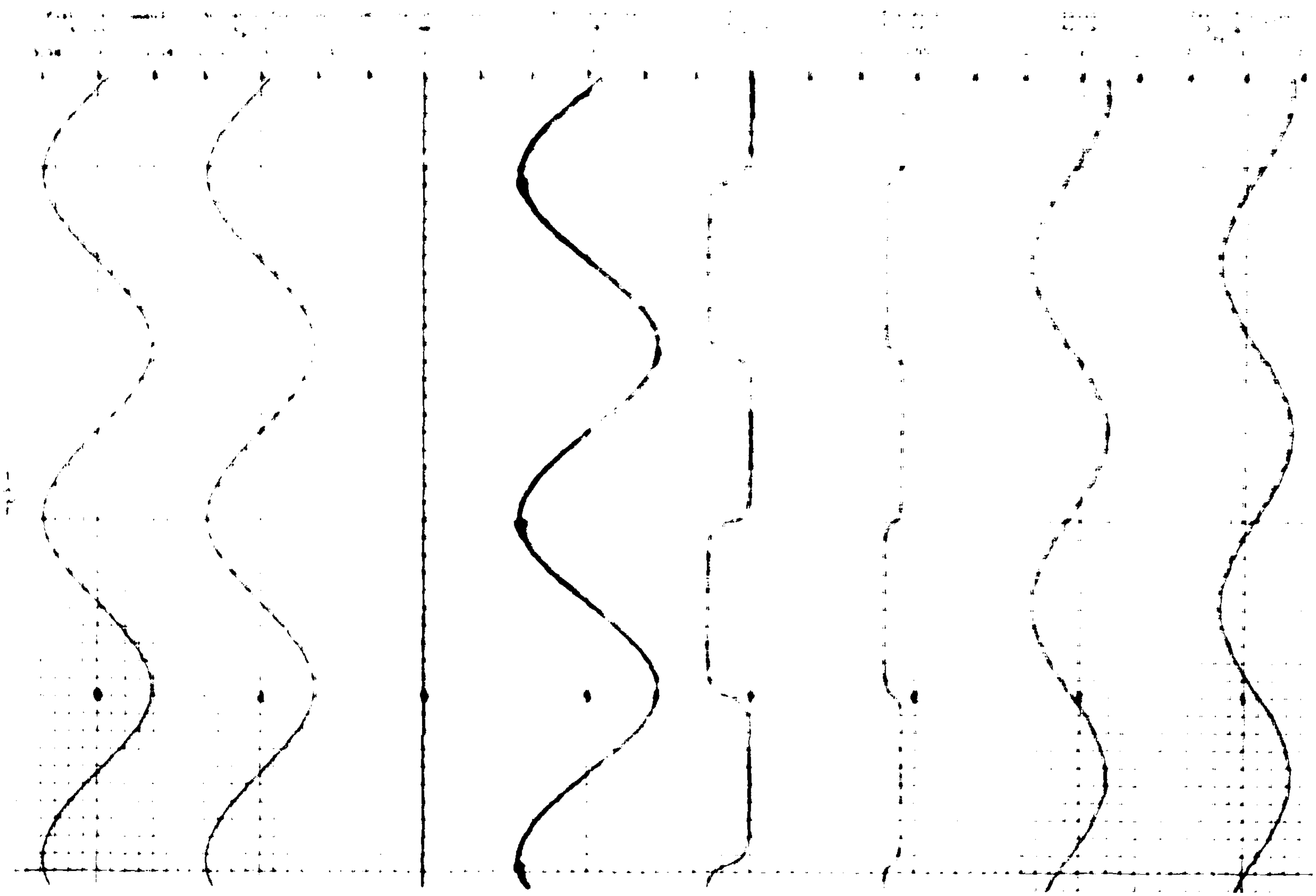


Figure 20. Full Scale Test, Commercial High Gain Valve, 0.05 Hz Sine Wave

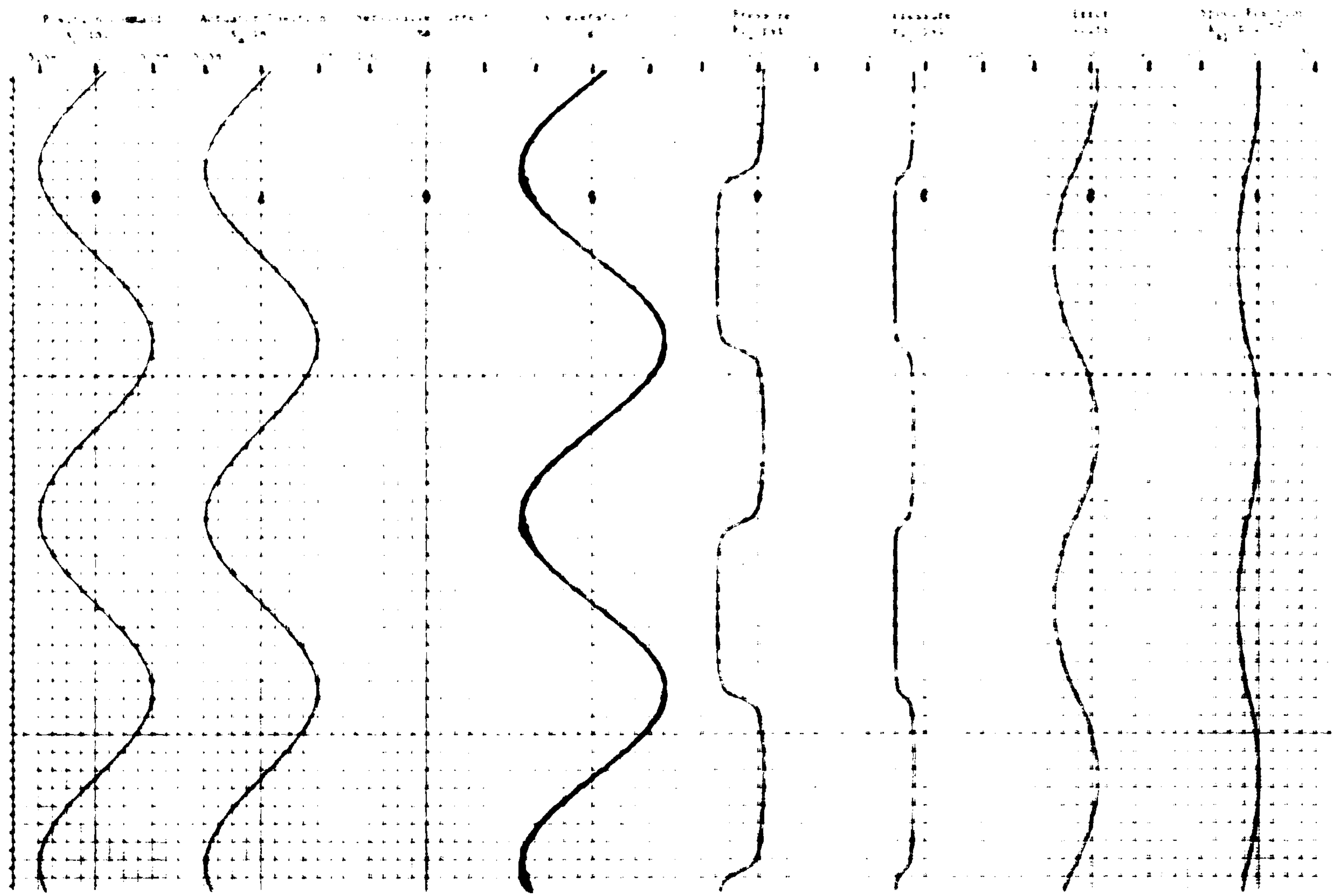


Figure 21. Full Scale Test, Franklin Low Gain Valve, 0.05 Hz Sine Wave

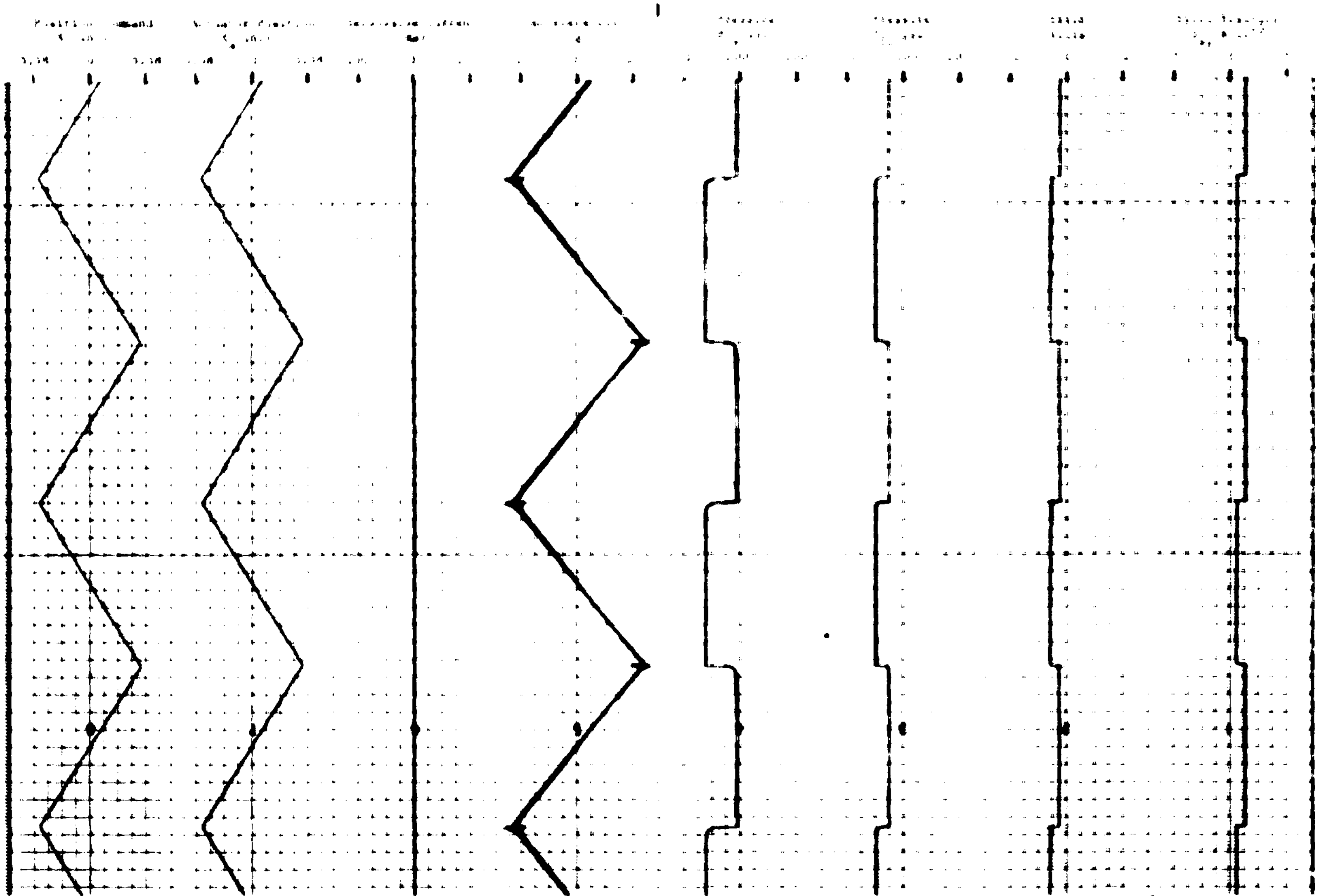


Figure 22. Full Scale Test, Commercial High Gain Valve,
Constant Velocity - 0.20 in/sec

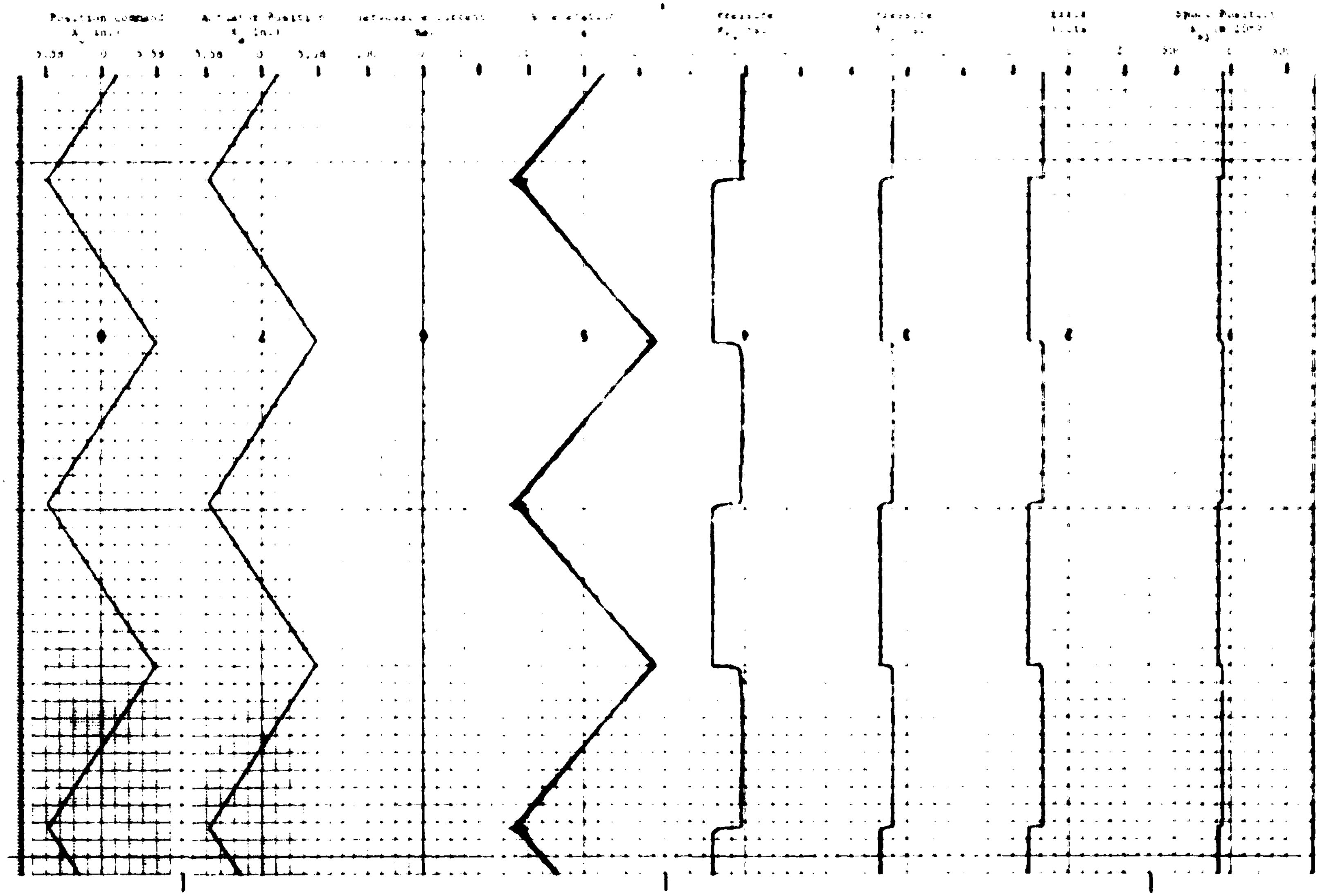


Figure 23. Full Scale Test, Franklin Low Gain Valve, Constant Velocity = 0.20 in/sec

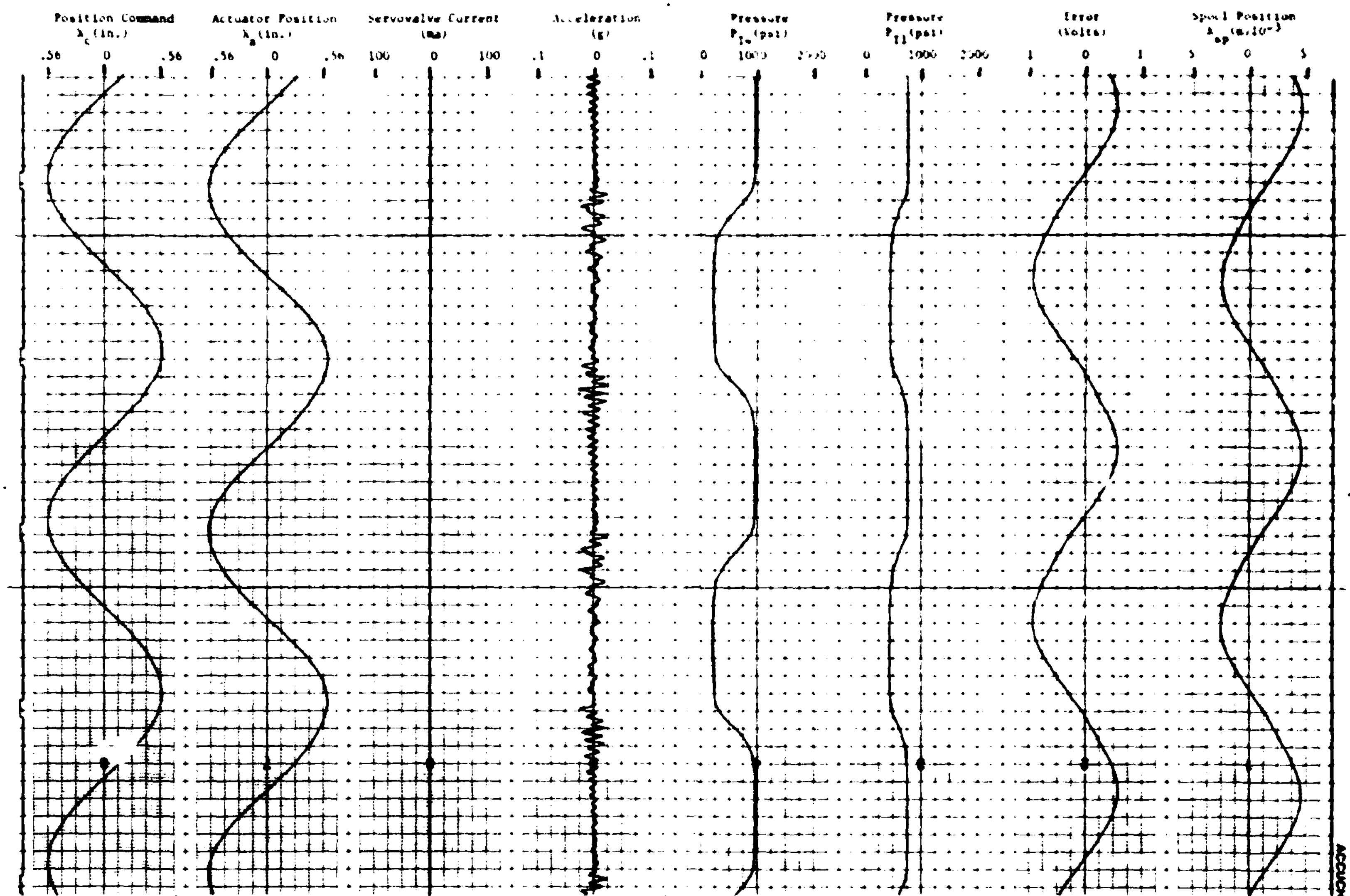


Figure 24. Full Scale Test, Commercial High Gain Valve
0.50 Hz Sine Wave

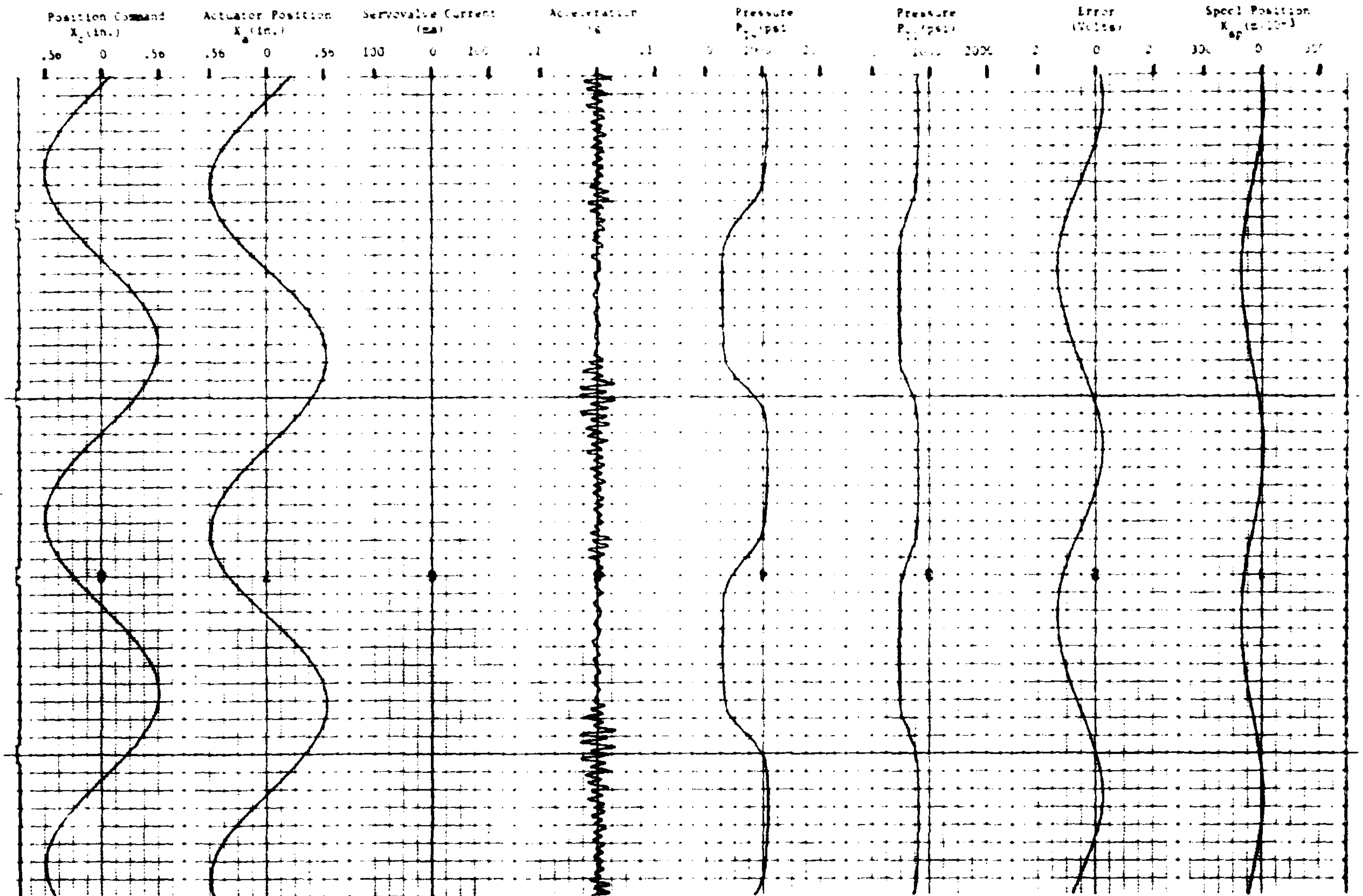


Figure 25. Full Scale Test, Franklin Low Gain Valve, 0.50 Hz Sine Wave

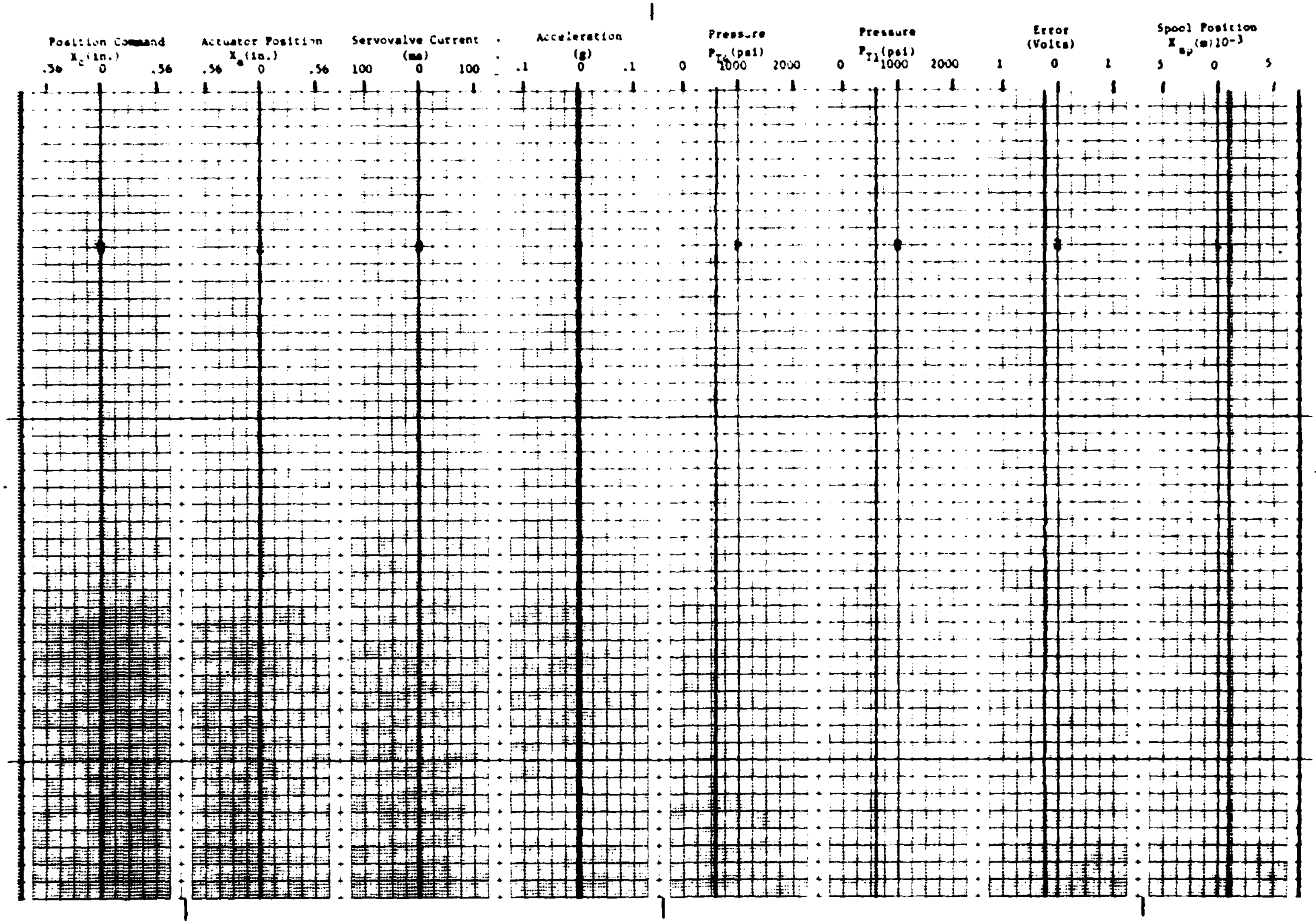


Figure 26. Full Scale Test, Commercial High Gain Valve, Zero Command Signal

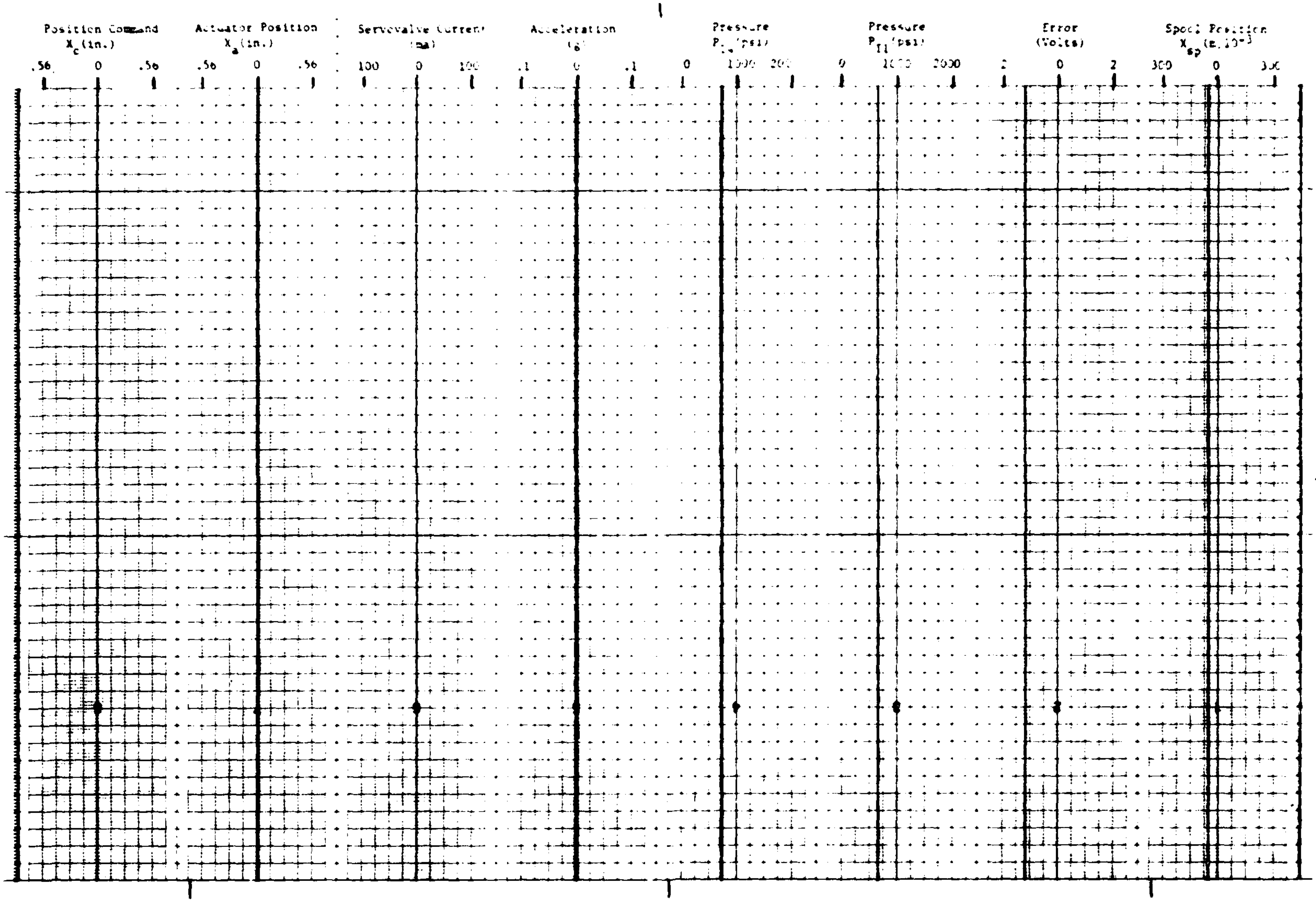
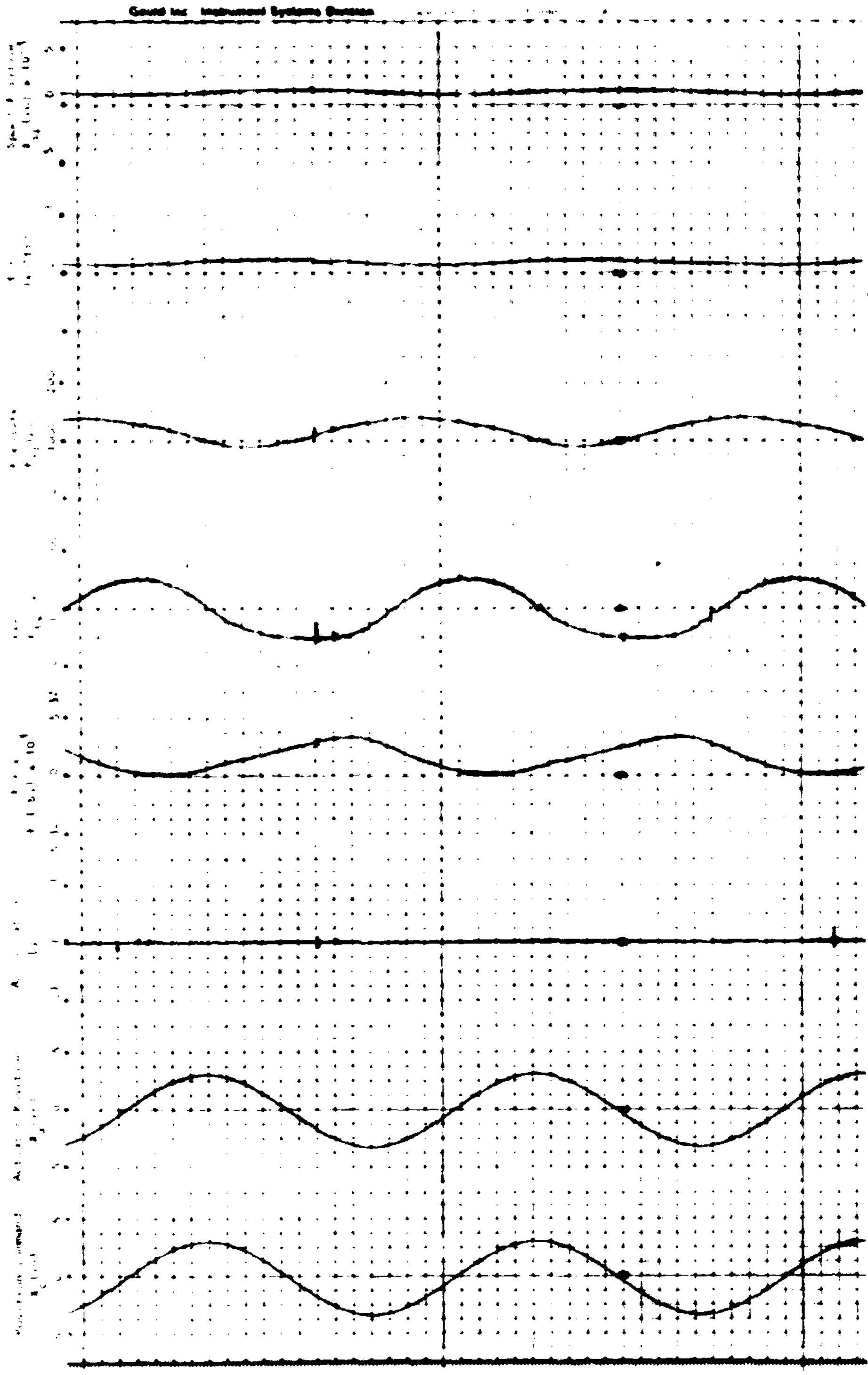


Figure 27. Full Scale Test, Franklin Low Gain Valve, Zero Command Signal

APPENDIX

A

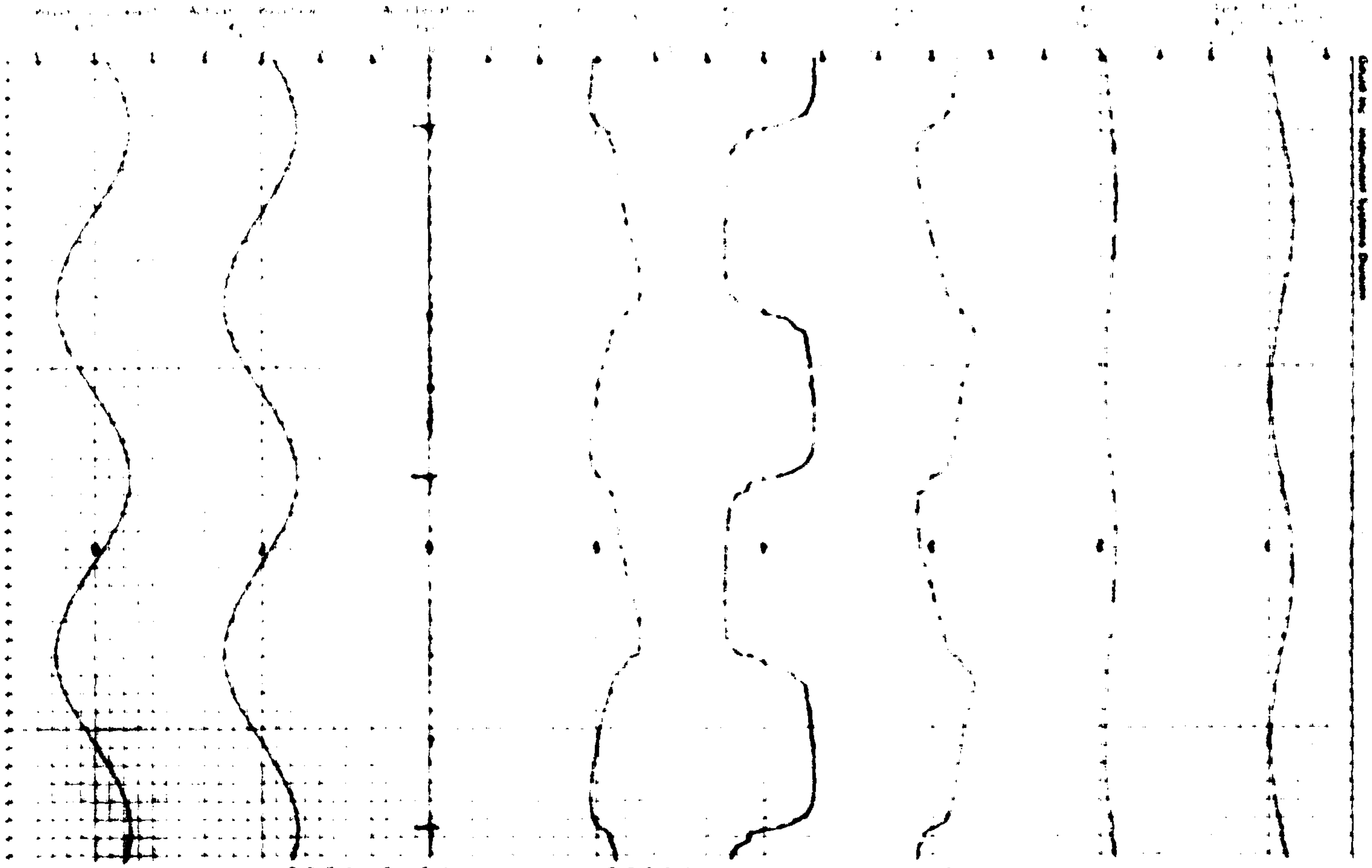
Small Scale System Tests,
Commercial High Gain Valve with Unequal Cylinder Areas



FREQUENCY: 0.01 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

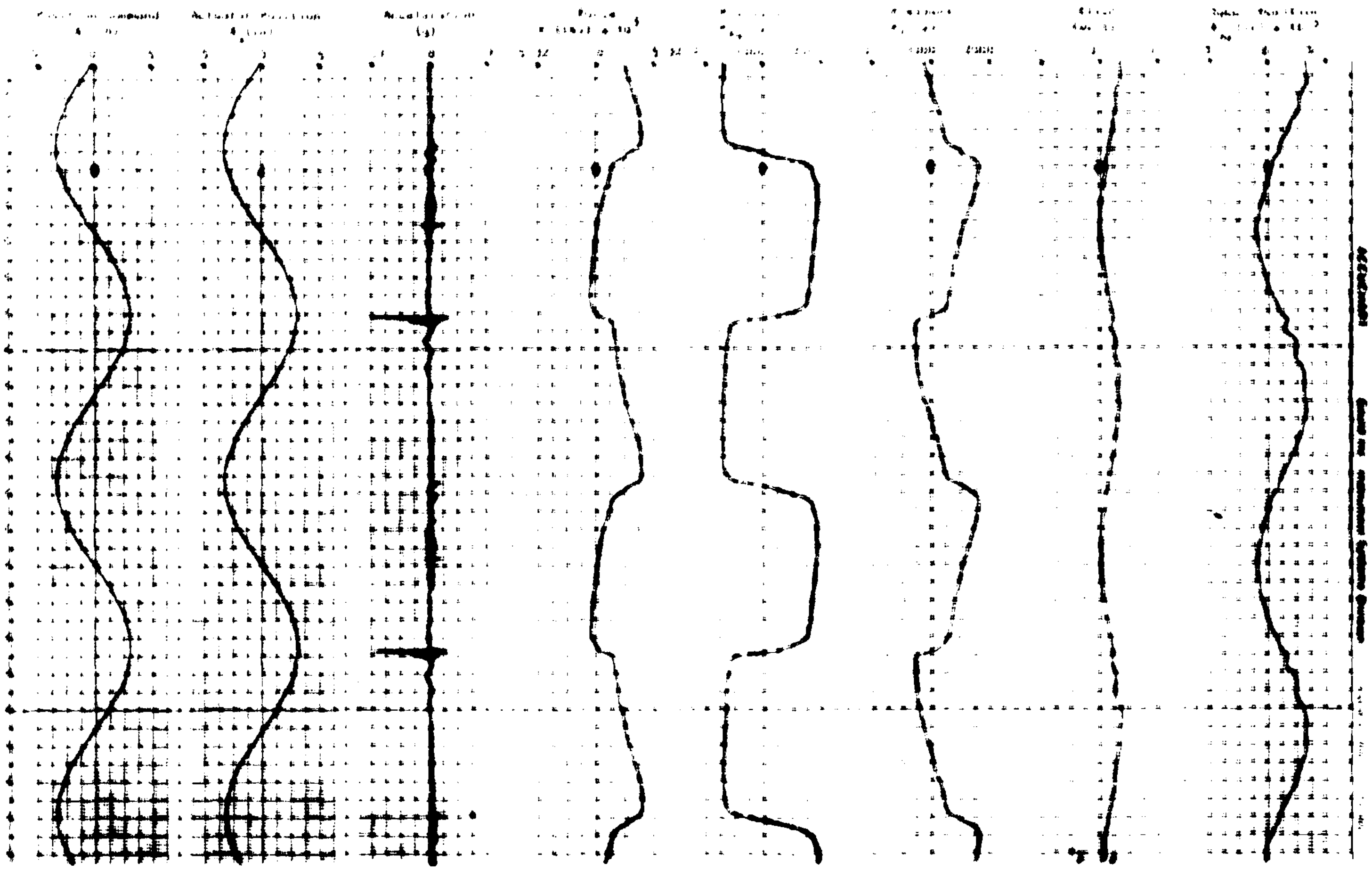
FIGURE: 4-1



FREQUENCY: [illegible] WAVEFORM: SINE LOAD: [illegible]

VALVE GAIN: HIGH CYLINDER AREA: [illegible]

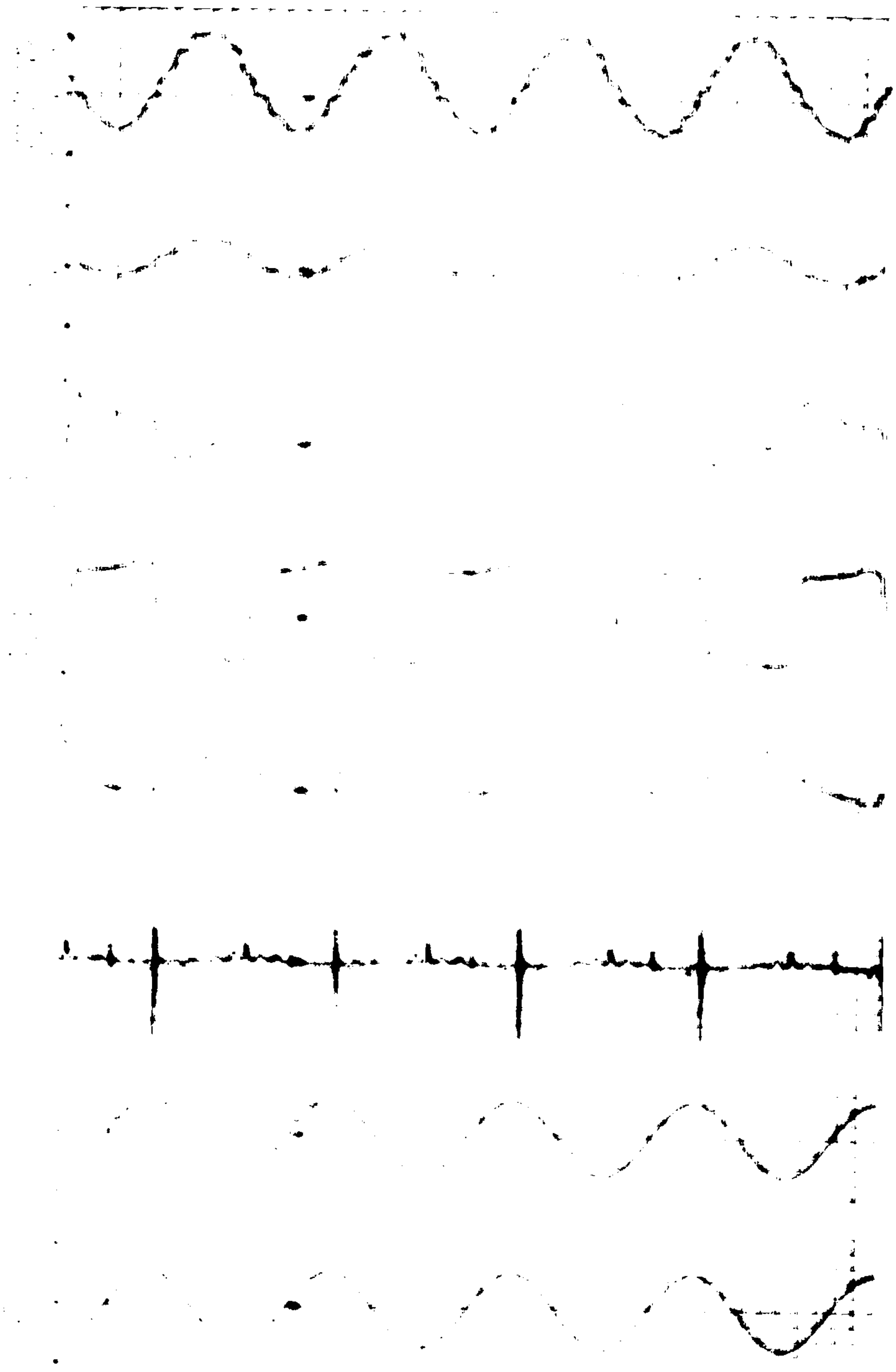
FIGURE: [illegible]

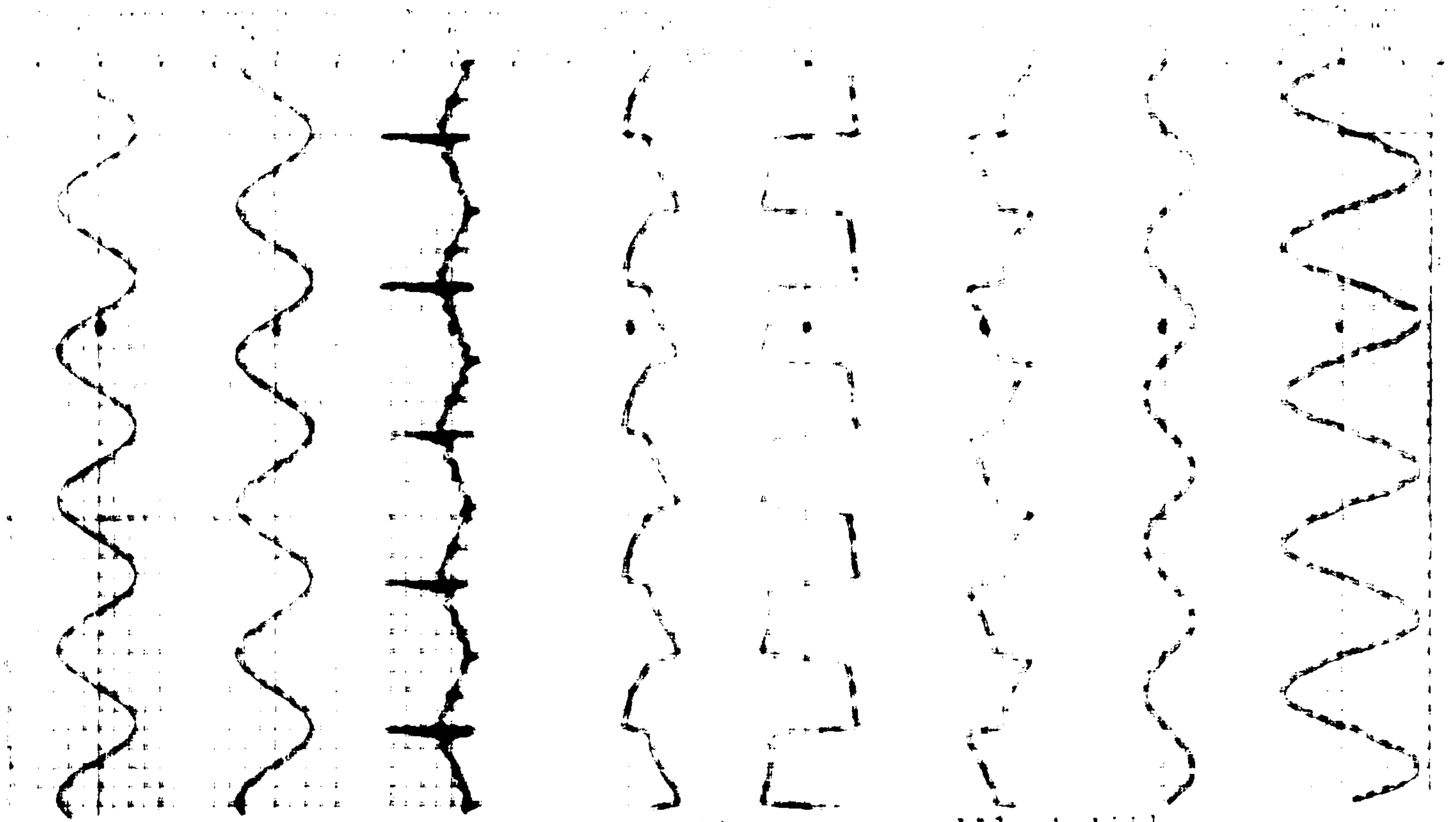


FREQUENCY: 0.10 WAVEFORM: SINE LOAD: [illegible]

VALVE GAIN: [illegible] CYLINDER AREA: [illegible]

FIGURE: A-3

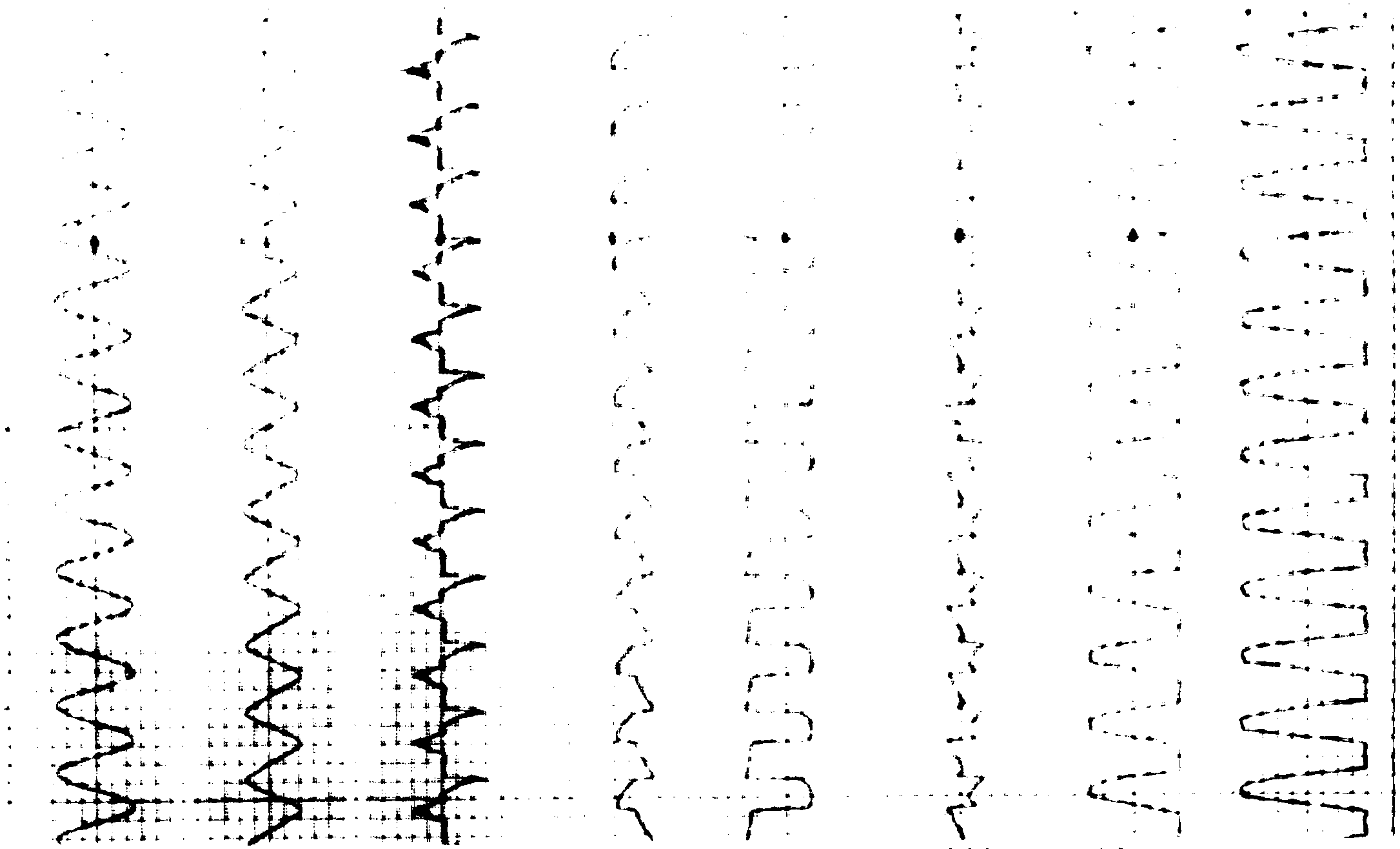




FREQUENCY [illegible] WAVEFORM [illegible] SIZE [illegible]

VALVE GAIN [illegible] CYLINDER AREA [illegible]

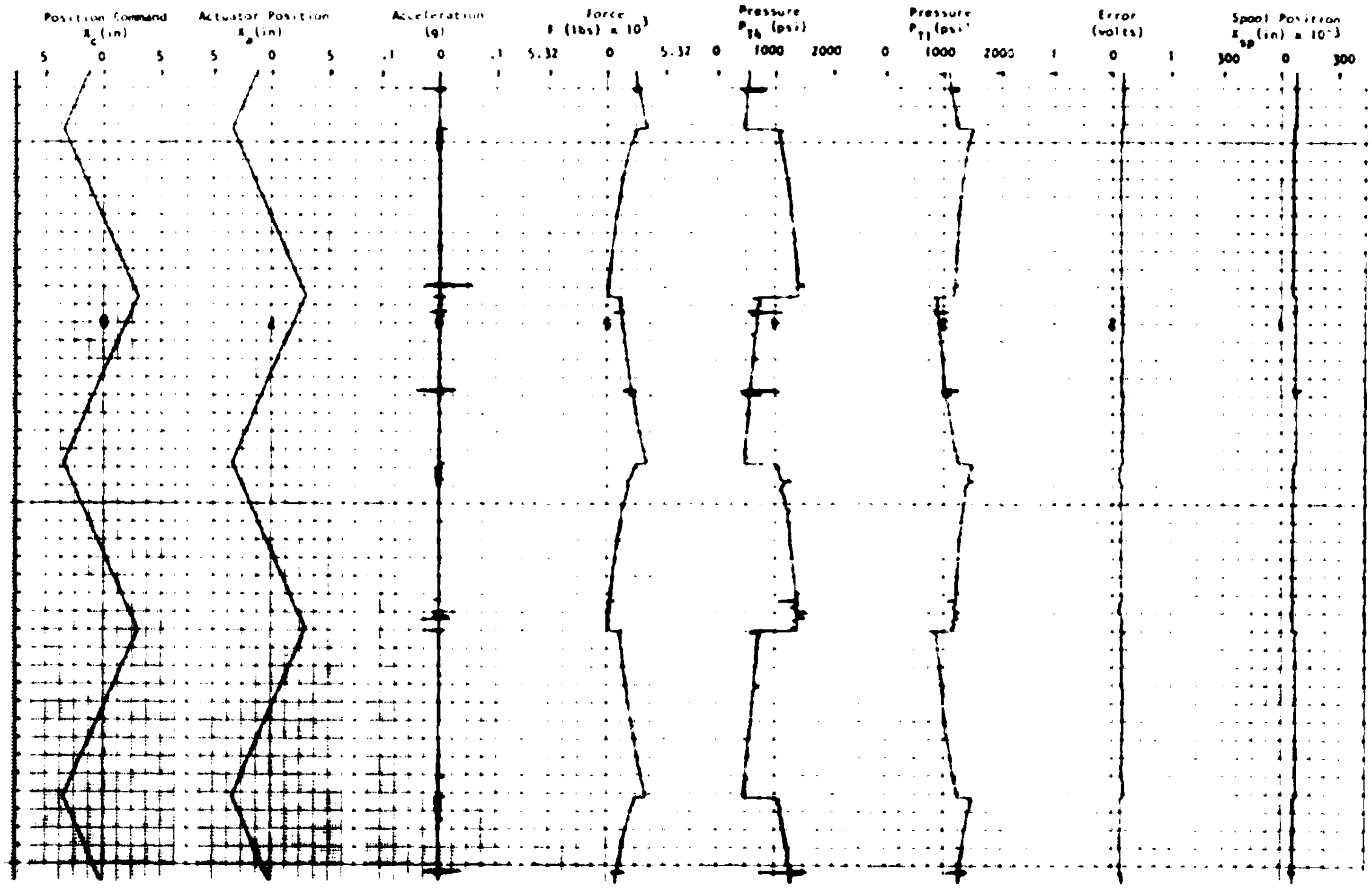
FIGURE [illegible]



FREQUENCY: [illegible] WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

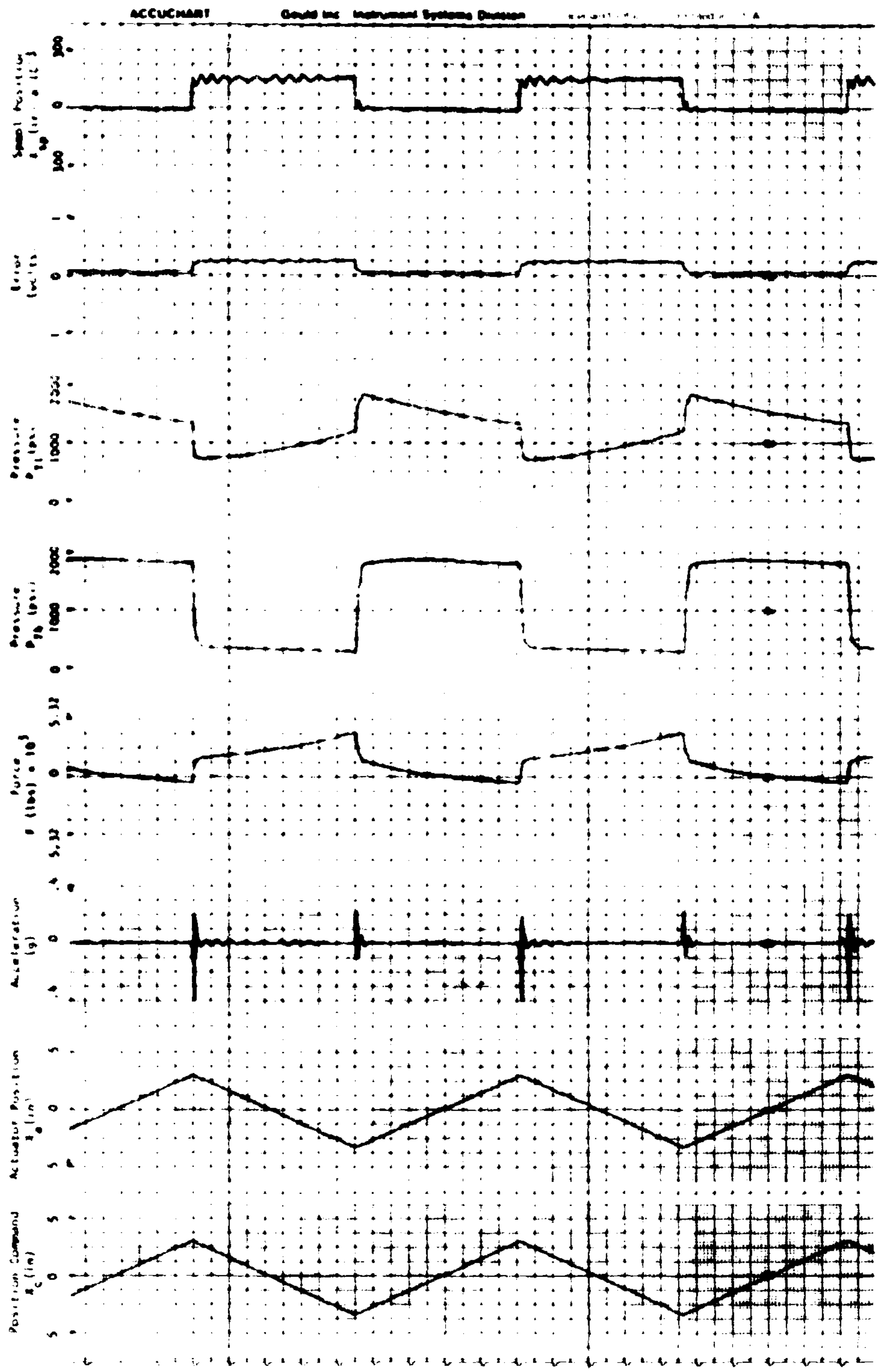
FIGURE: A-6



FREQUENCY: 0.01 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

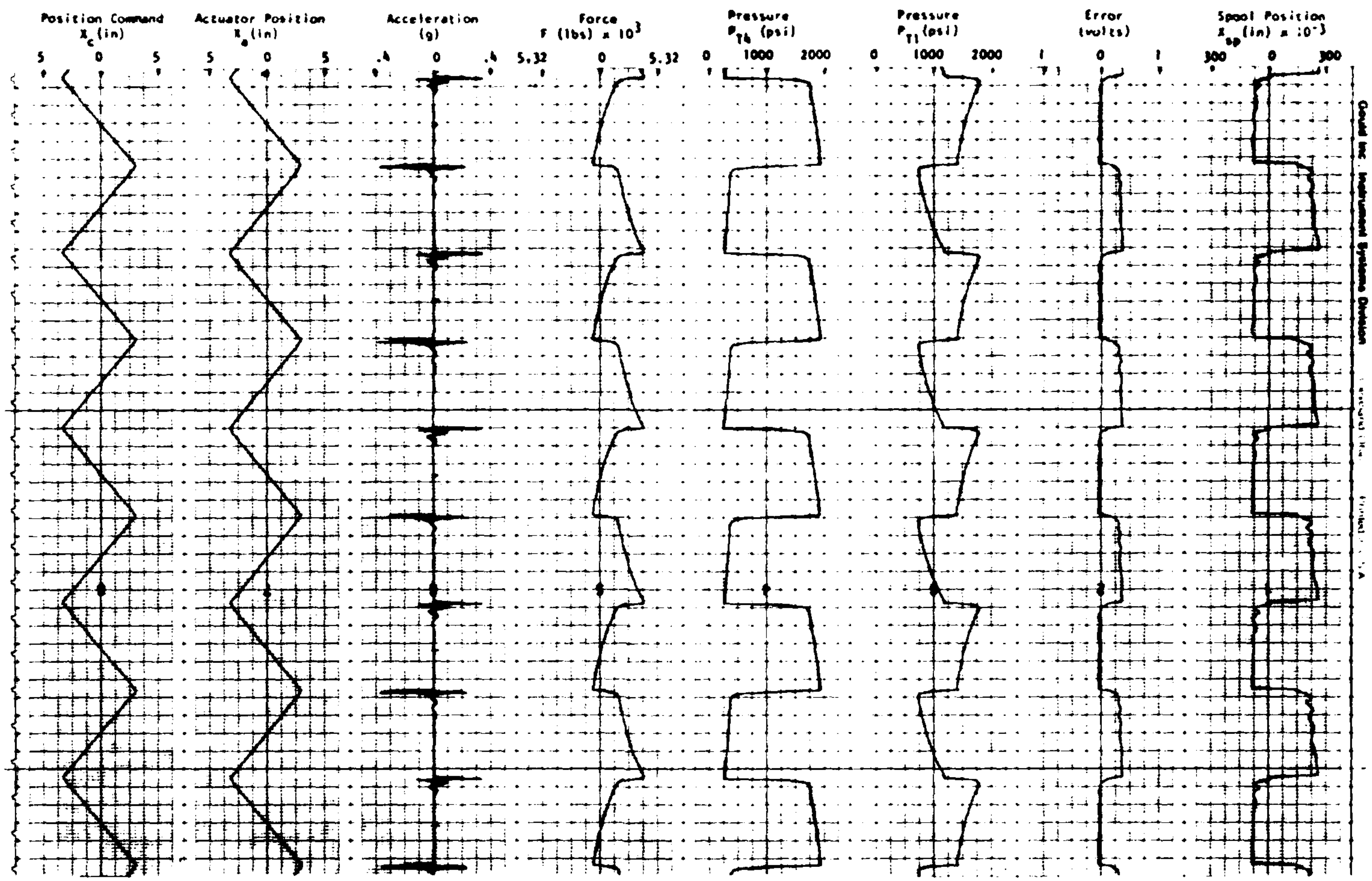
FIGURE: A-7



FREQUENCY: 0.10 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

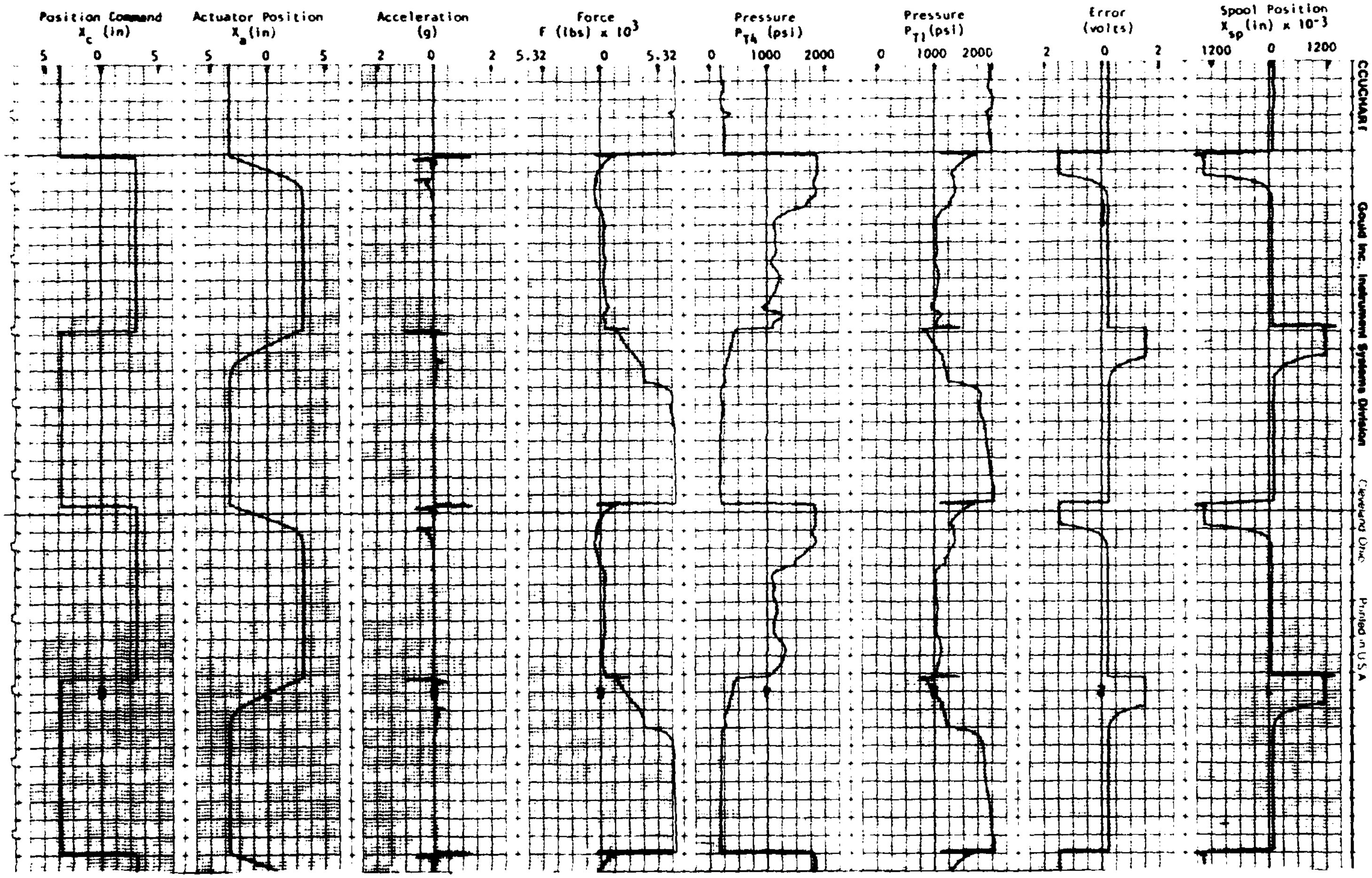
FIGURE: A-8



FREQUENCY: 0.20 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

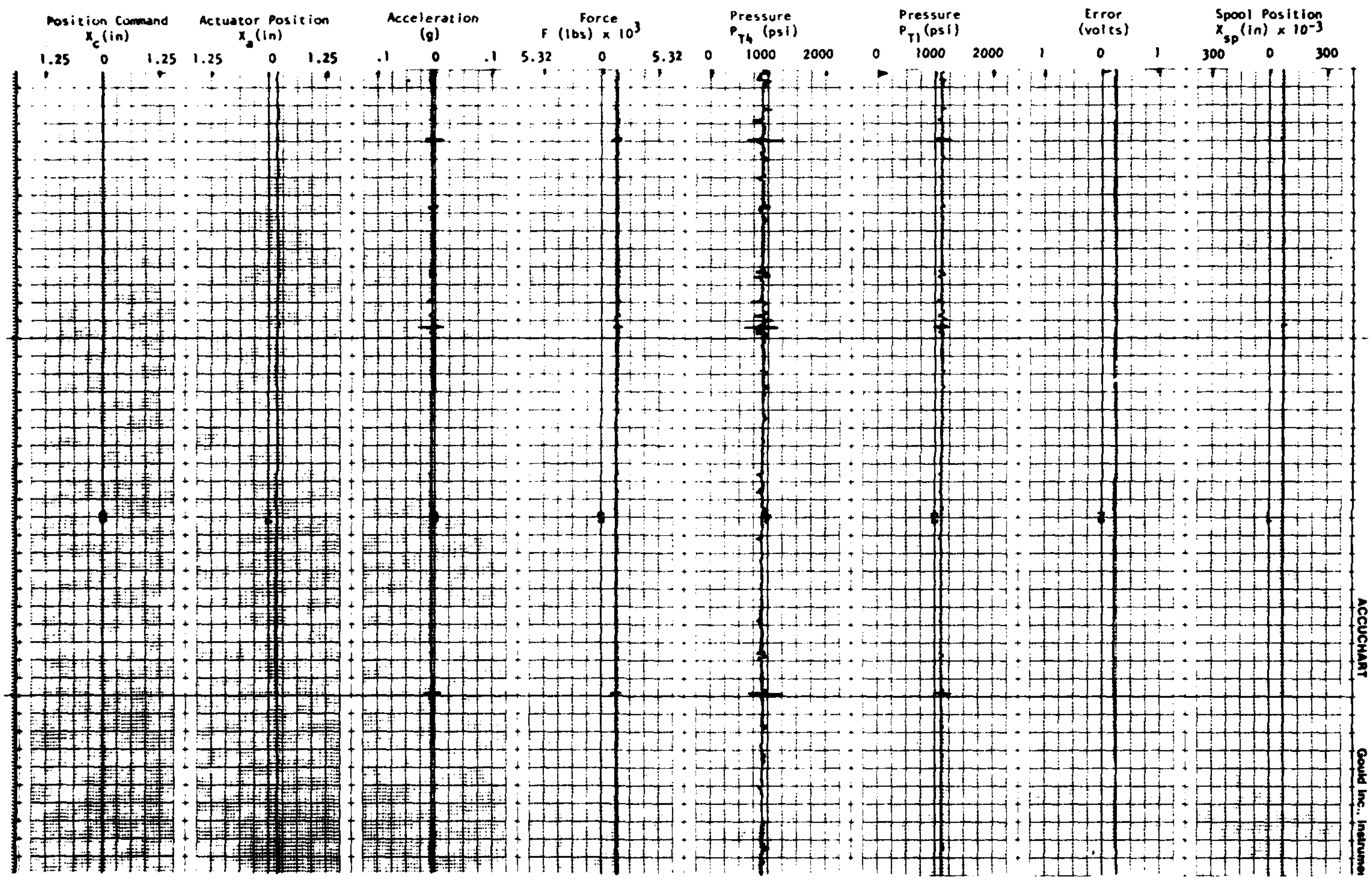
FIGURE: A-9



FREQUENCY: 0.20 WAVEFORM: SQUARE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

FIGURE: A-10



FREQUENCY: 0.00 WAVEFORM: ZERO LOAD: BIASED

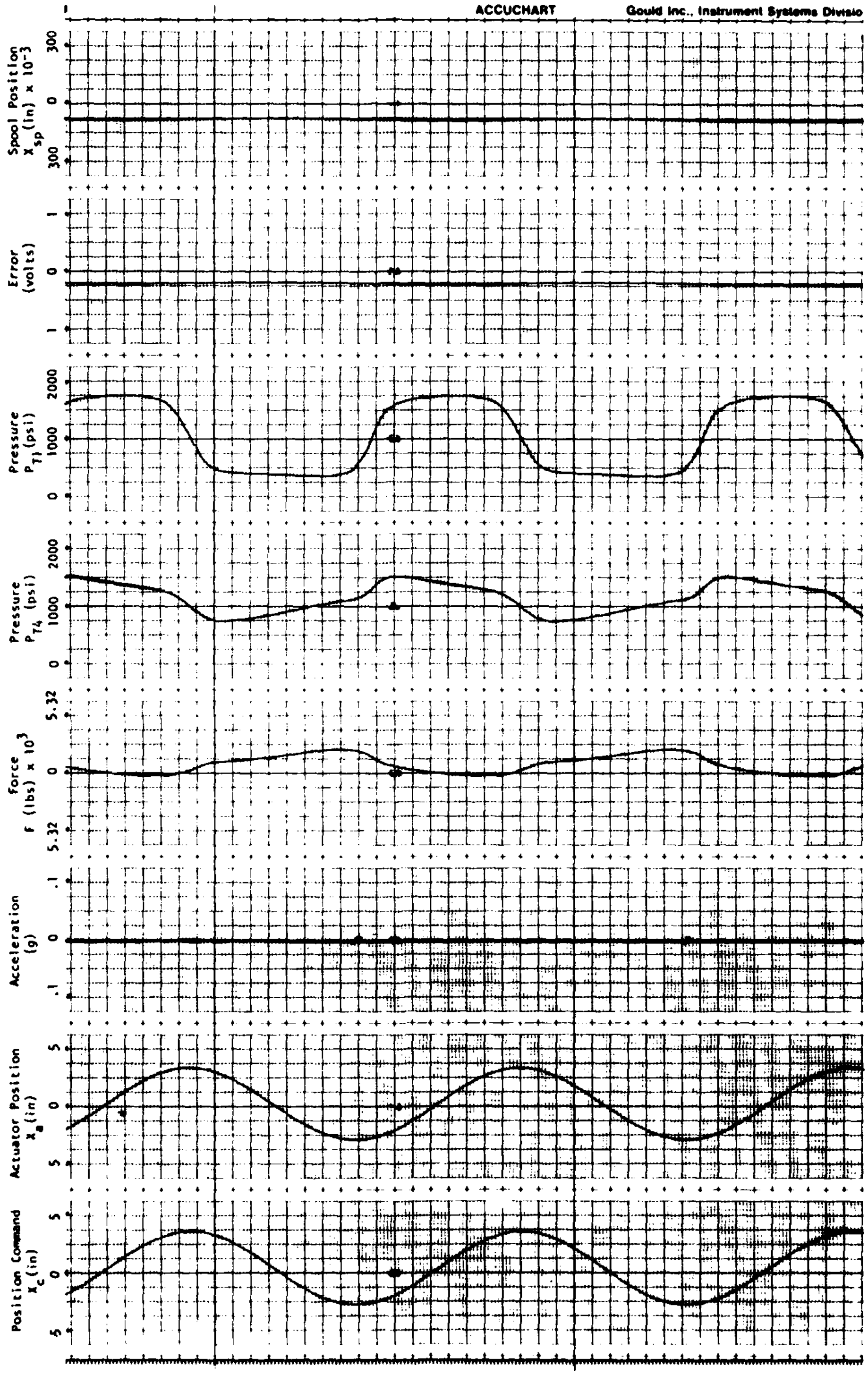
VALVE GAIN: HIGH CYLINDER AREA: UNEQUAL

FIGURE: A-11

APPENDIX

B

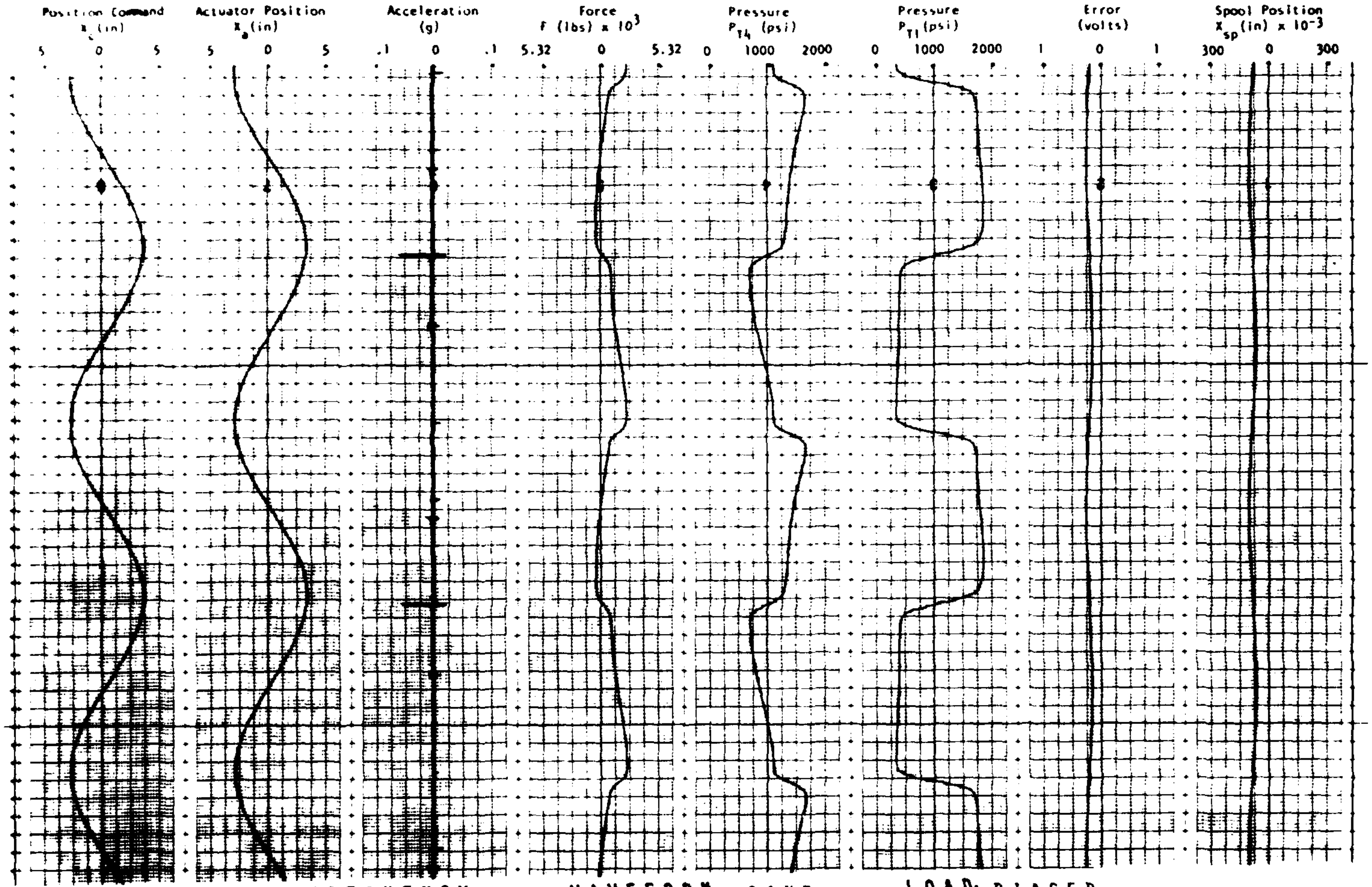
Small Scale System Tests,
Franklin Low Gain Valve with Unequal Cylinder Areas



FREQUENCY: 0.01 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

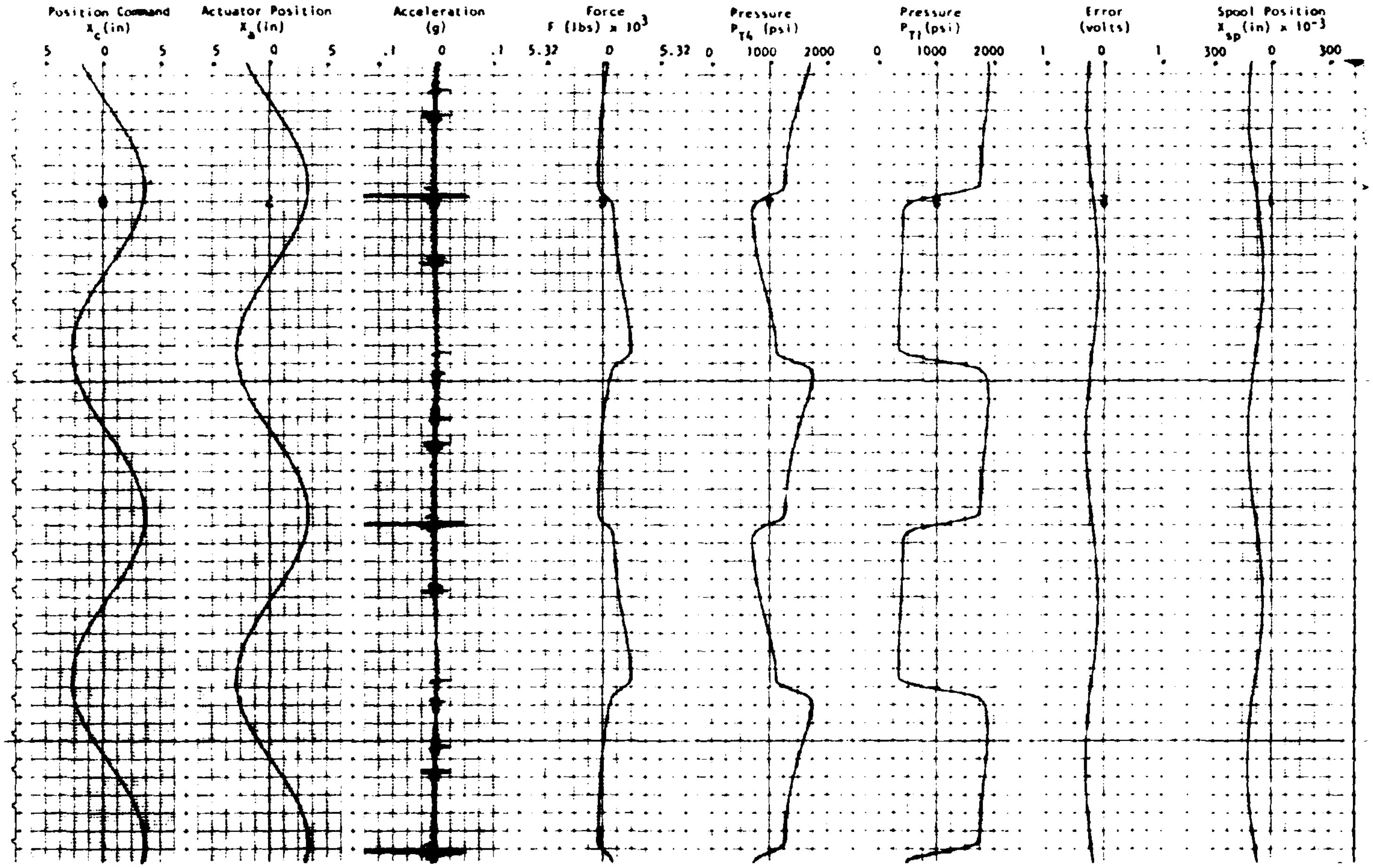
FIGURE: B-1



FREQUENCY: 0.05 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

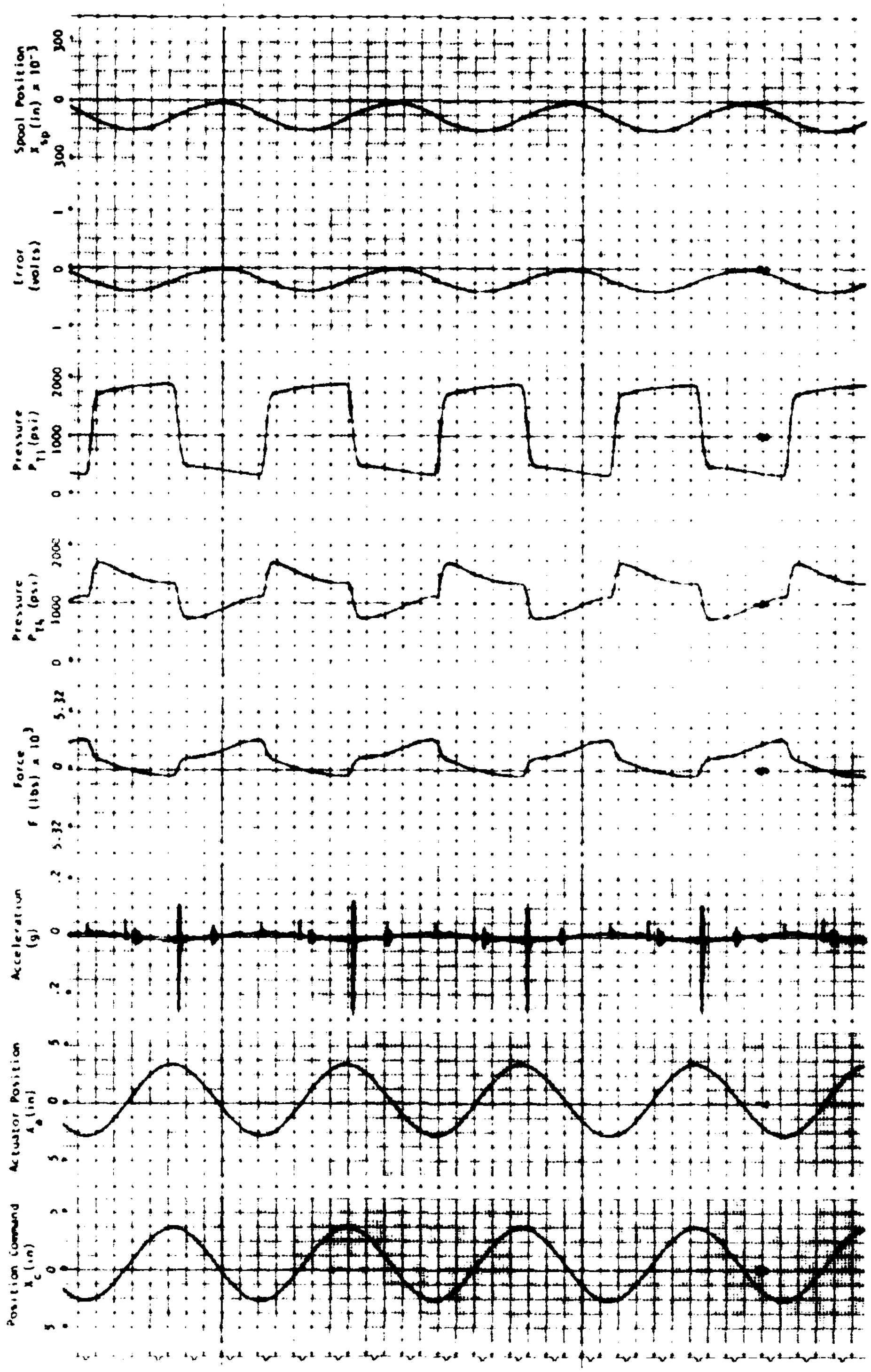
FIGURE: B-2



FREQUENCY: 0.10 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

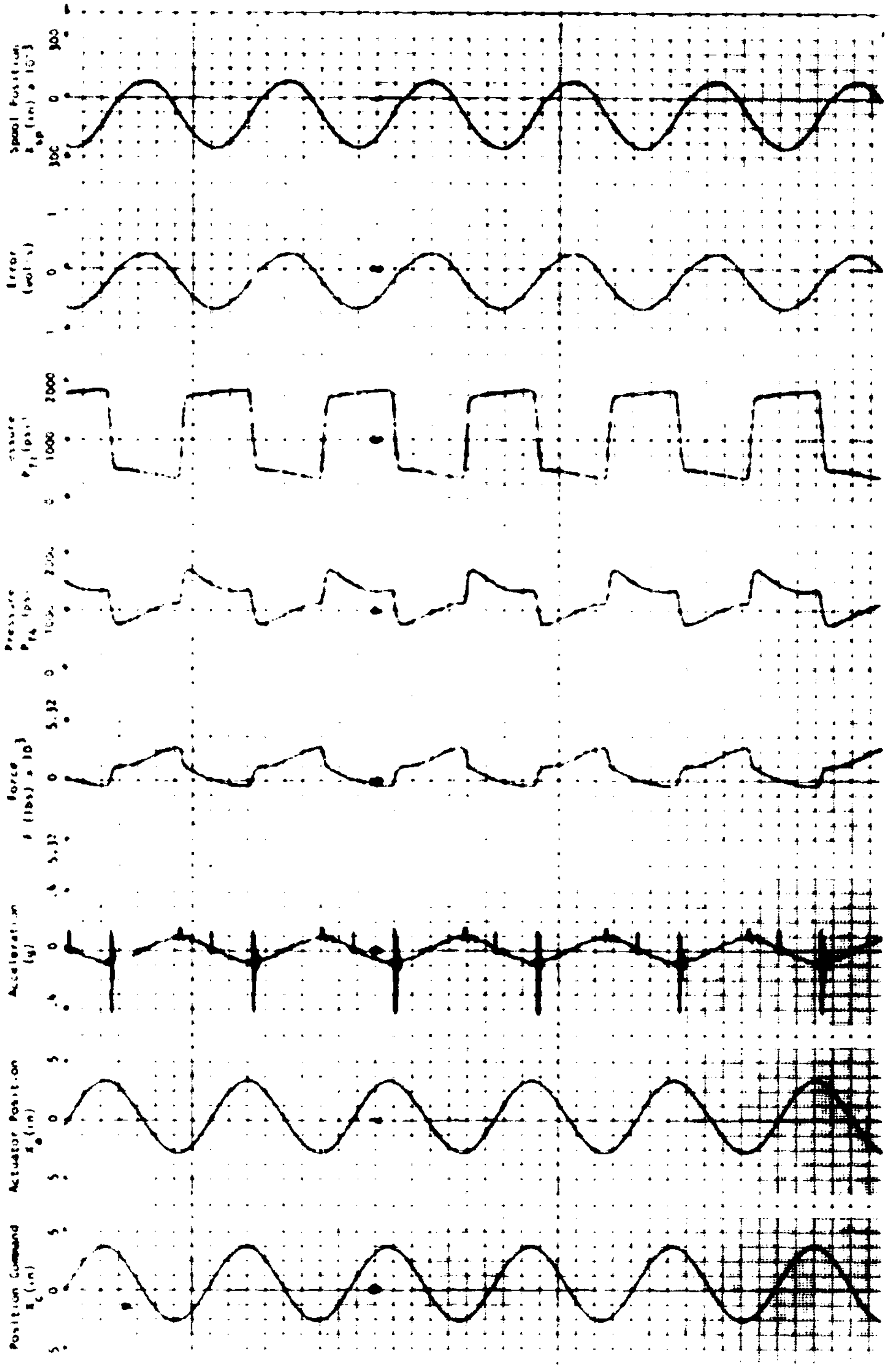
FIGURE: B-3



FREQUENCY: 0.20 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

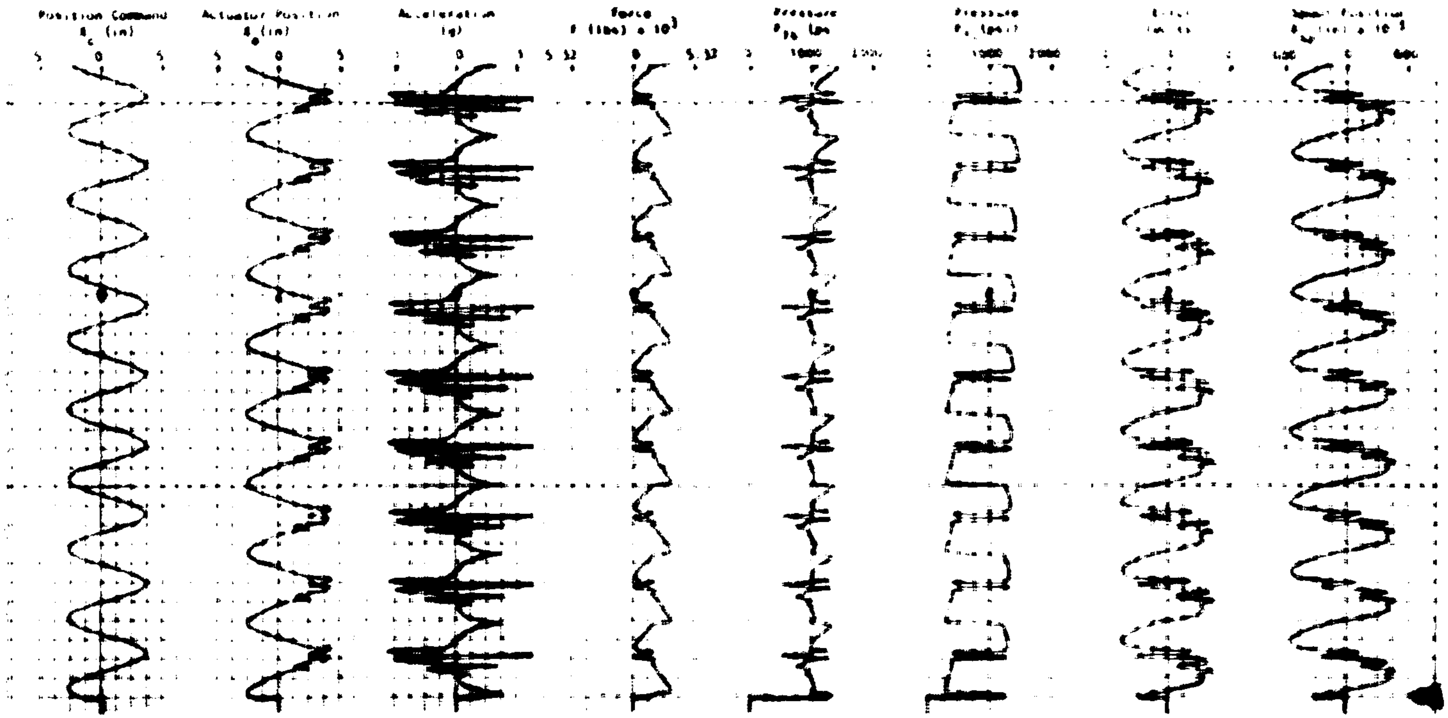
FIGURE: B-4



FREQUENCY: 0.50 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

FIGURE: B-5



FREQUENCY: 1.00 WAVEFORM: [illegible] LOAD: [illegible]

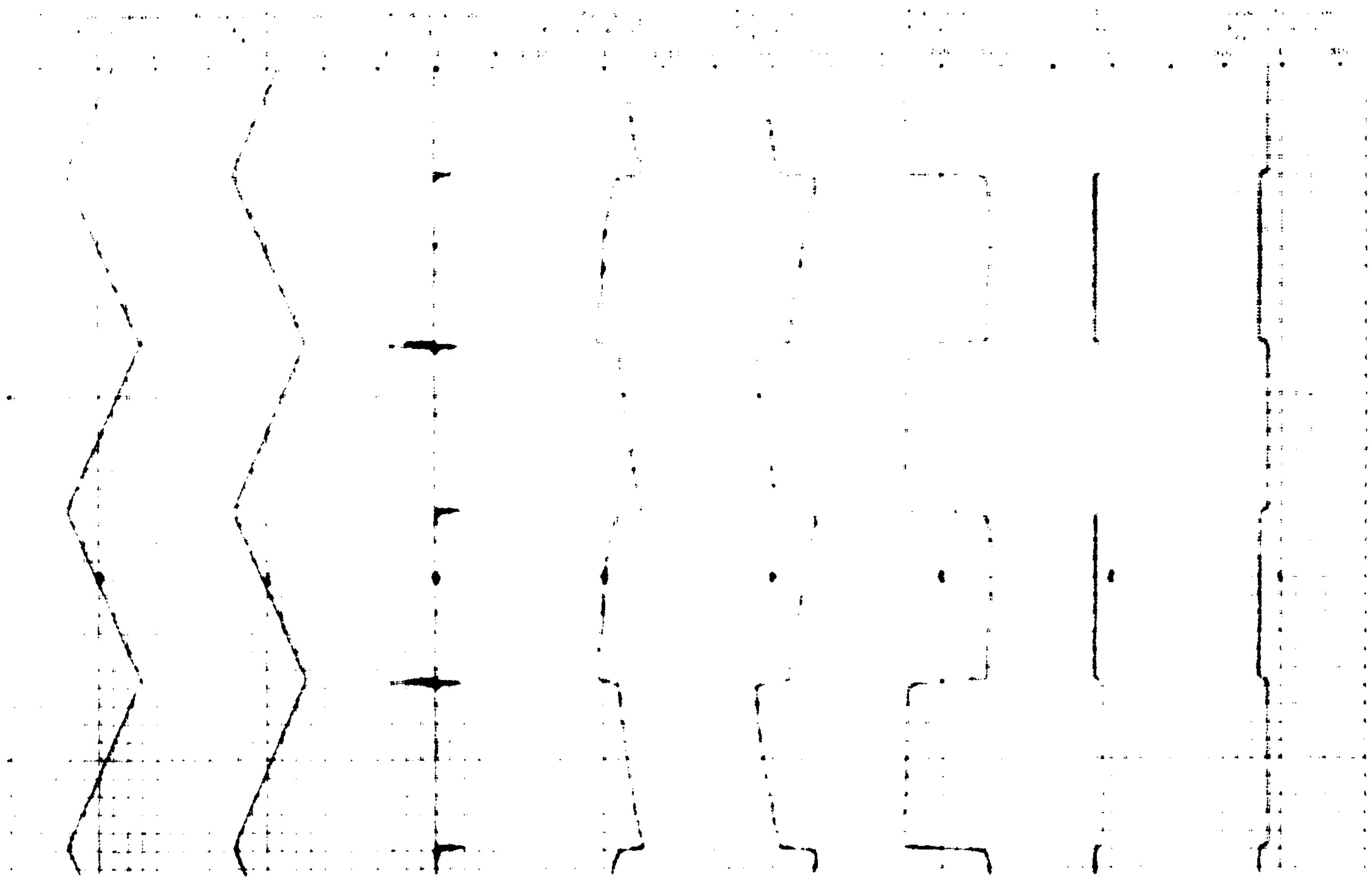
VALVE GAIN: [illegible] CYLINDER ASSY [illegible]

FIGURE: [illegible]

FREQUENCY [illegible] WAVEFORM [illegible] LOAD [illegible]

VALVE GAIN [illegible] CYLINDER AREA [illegible]

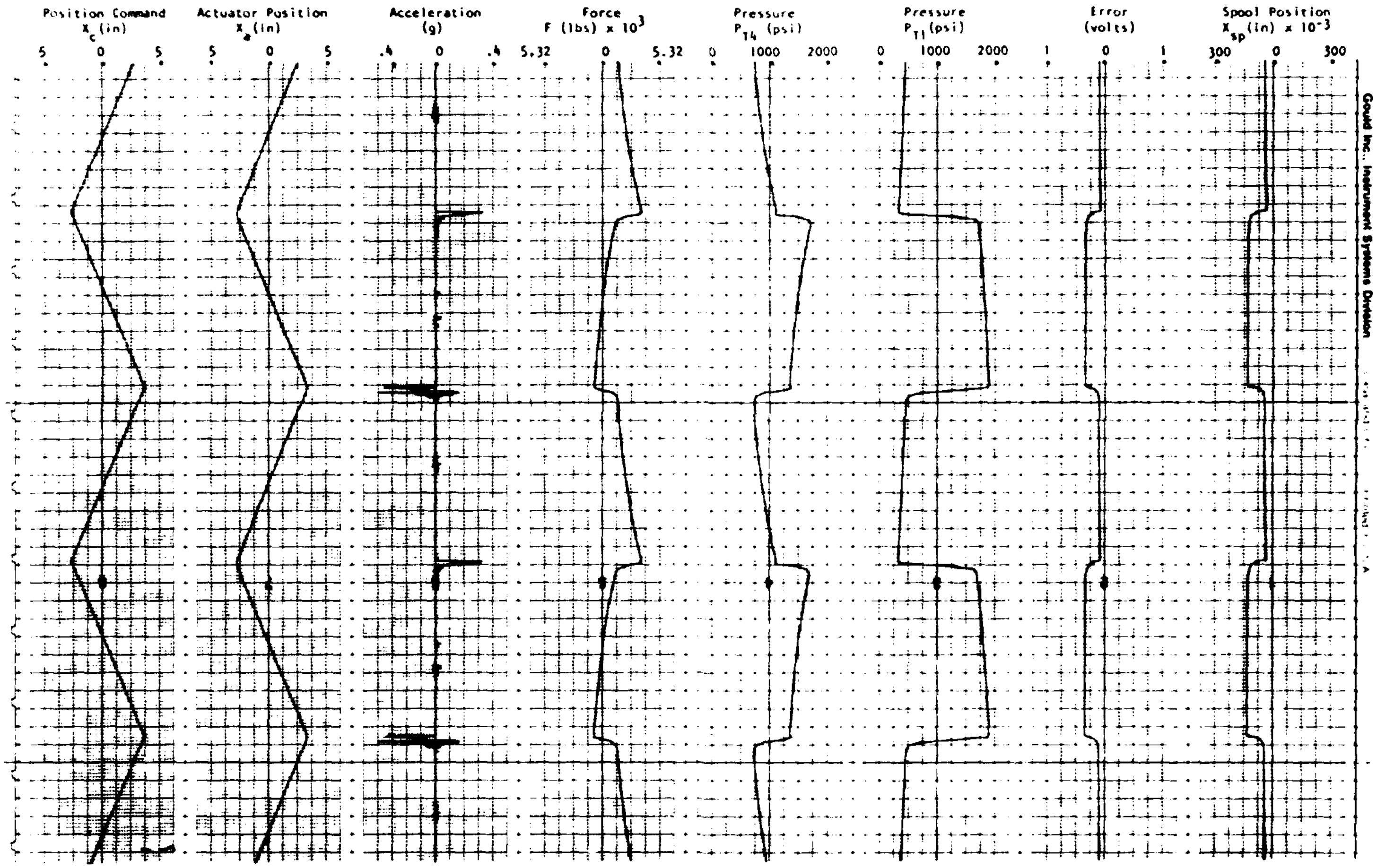
FIGURE [illegible]



FREQUENCY: ____ WAVEFORM: ____ LOAD: ____

VALVE GAIN: ____ CYLINDER AREA: ____

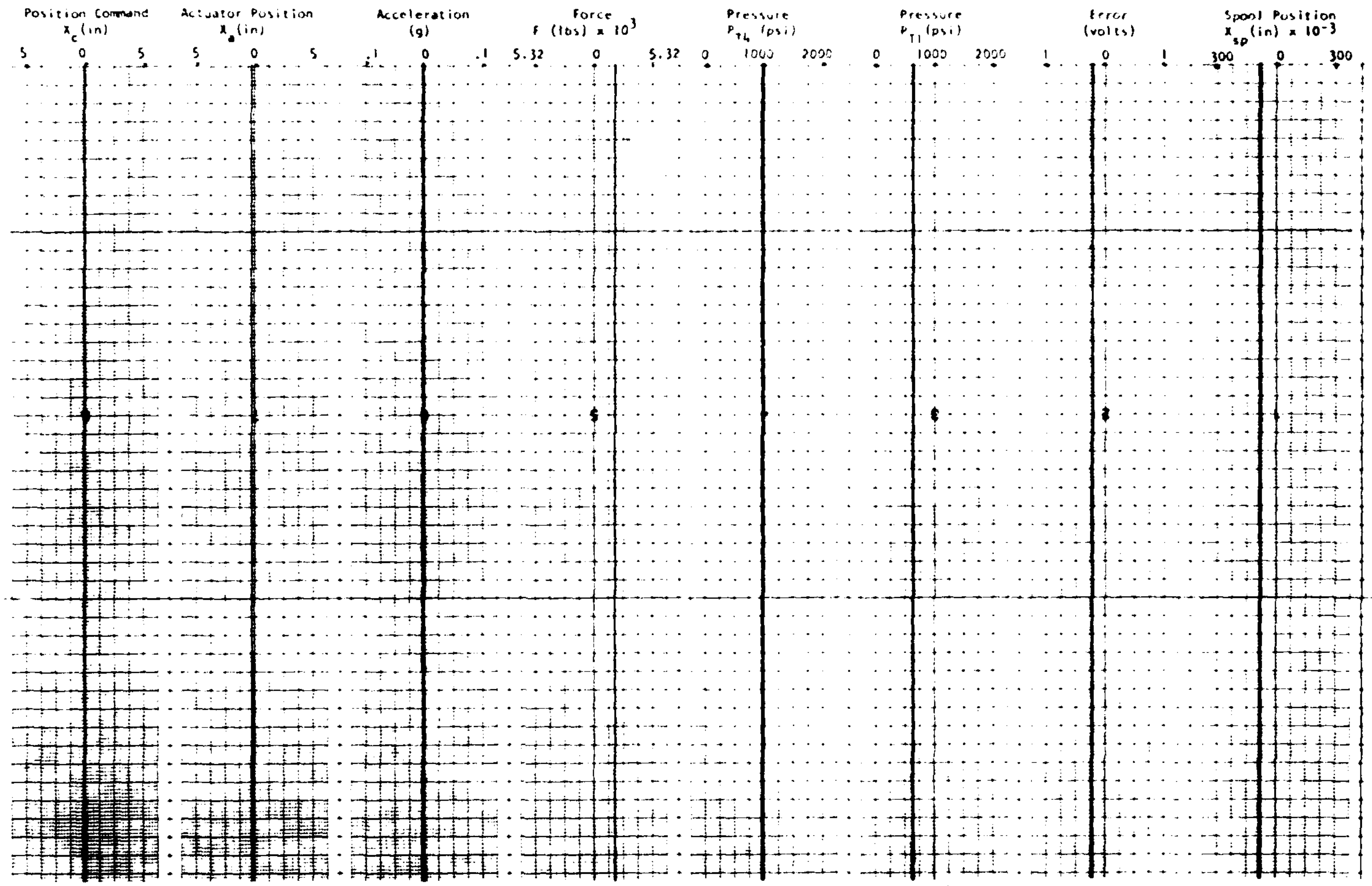
FIGURE: ____



FREQUENCY: 0.20 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

FIGURE: B-9



FREQUENCY: 0.00 WAVEFORM: D.C. LOAD: BIASED

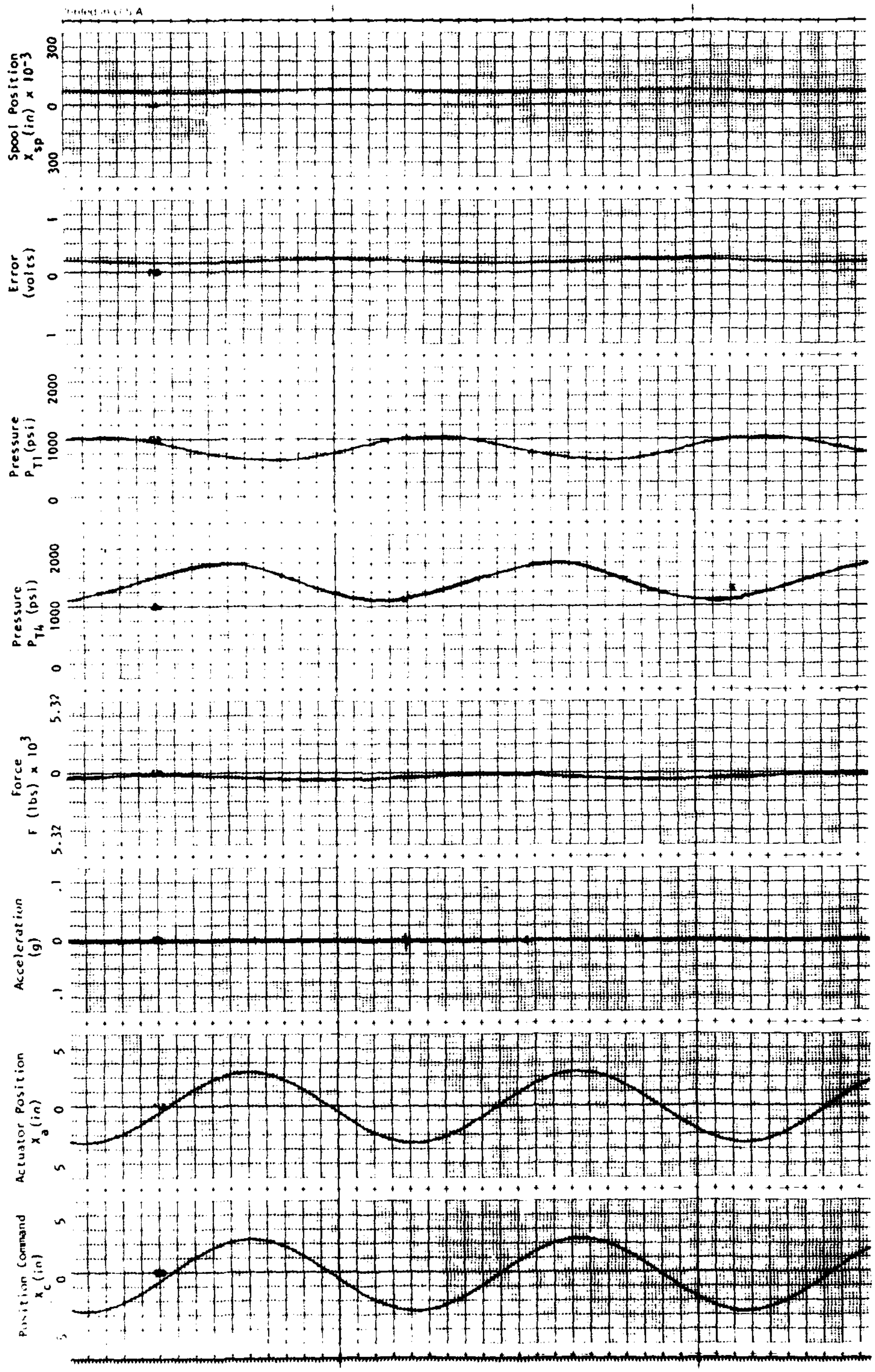
VALVE GAIN: LOW CYLINDER AREA: UNEQUAL

FIGURE: B-10

APPENDIX

C

Small Scale System Tests,
Commercial High Gain Valve with Equal Cylinder Areas



FREQUENCY: 0.01 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: EQUAL

FIGURE: C-1

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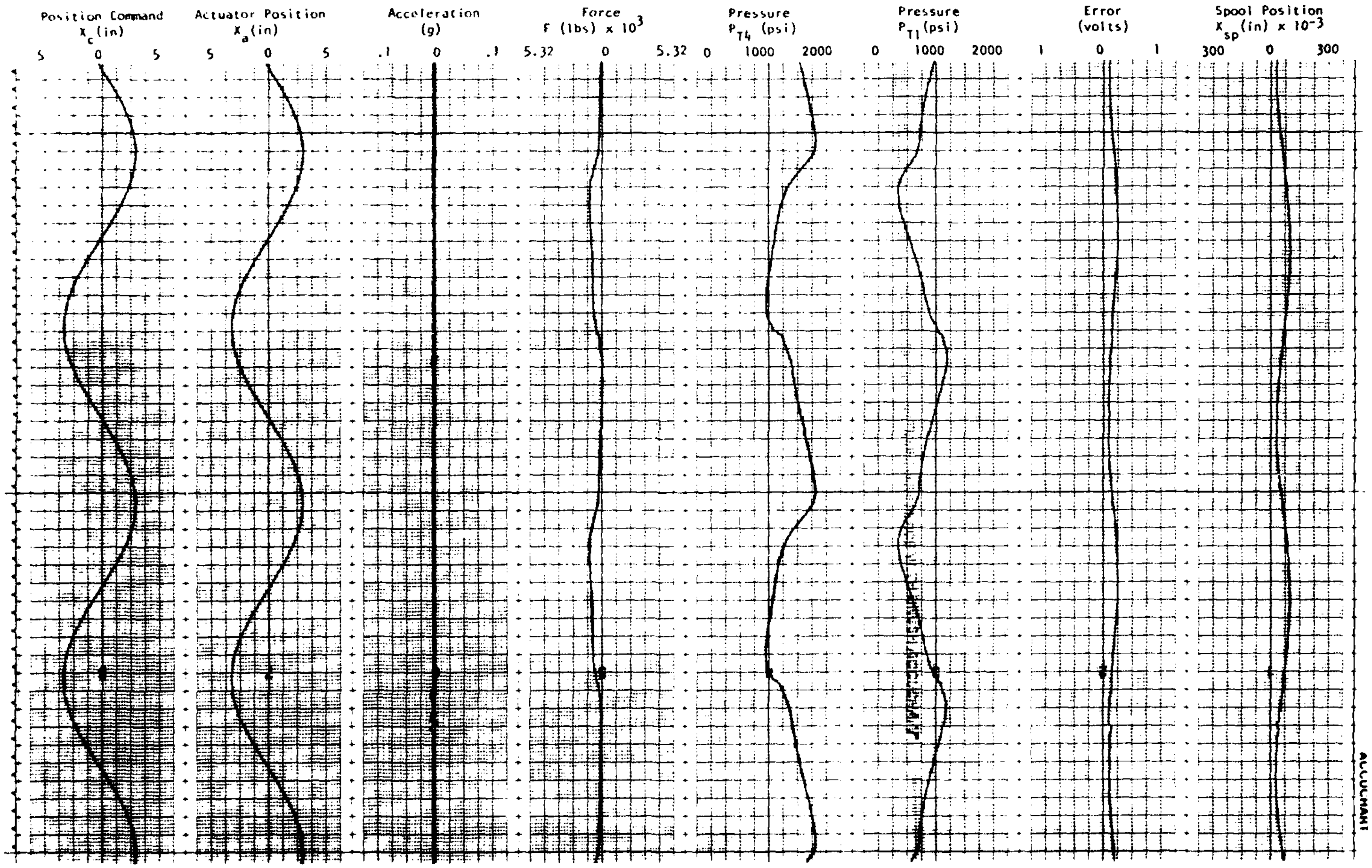
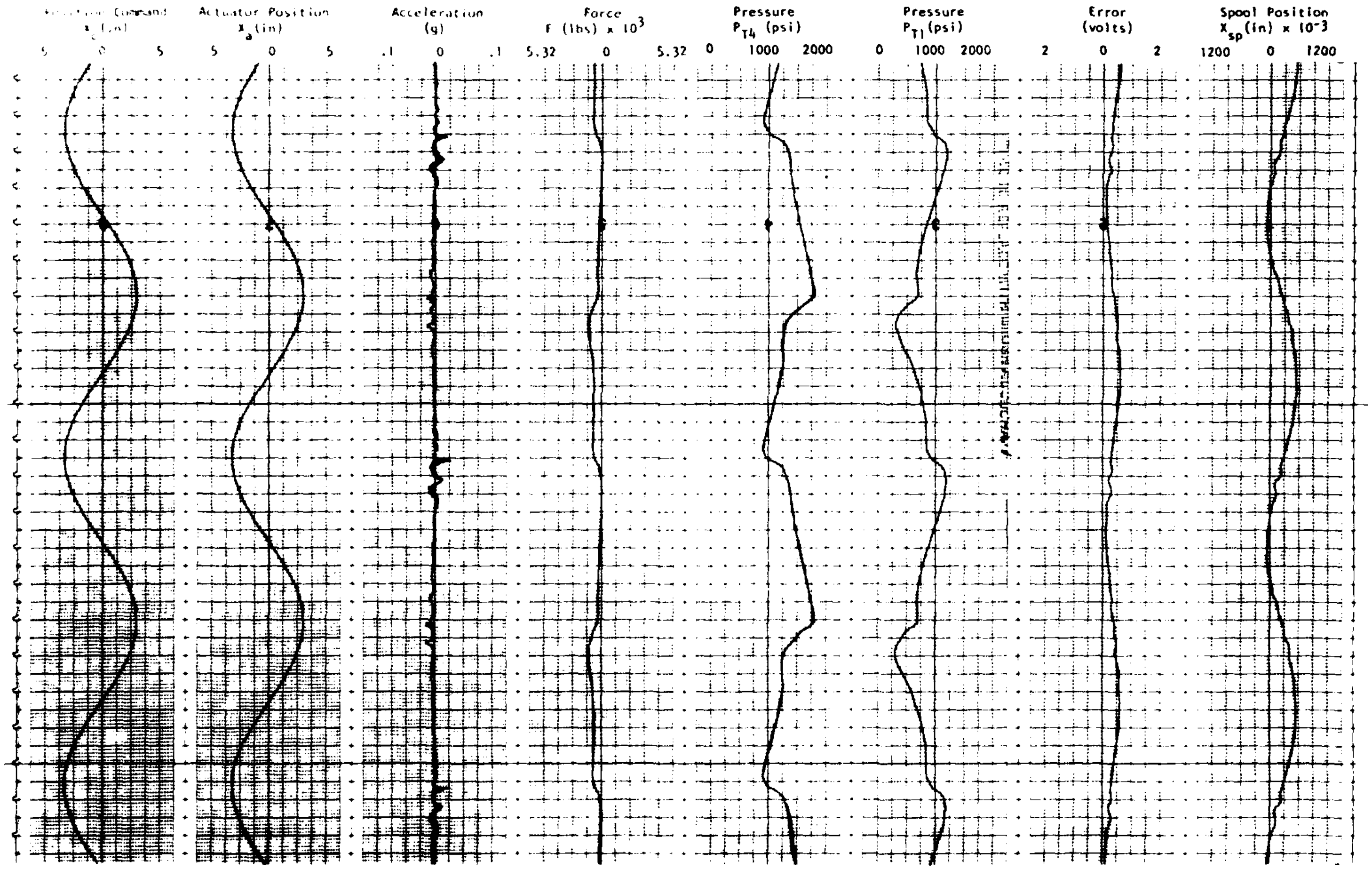


FREQUENCY: 0.05 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: EQUAL

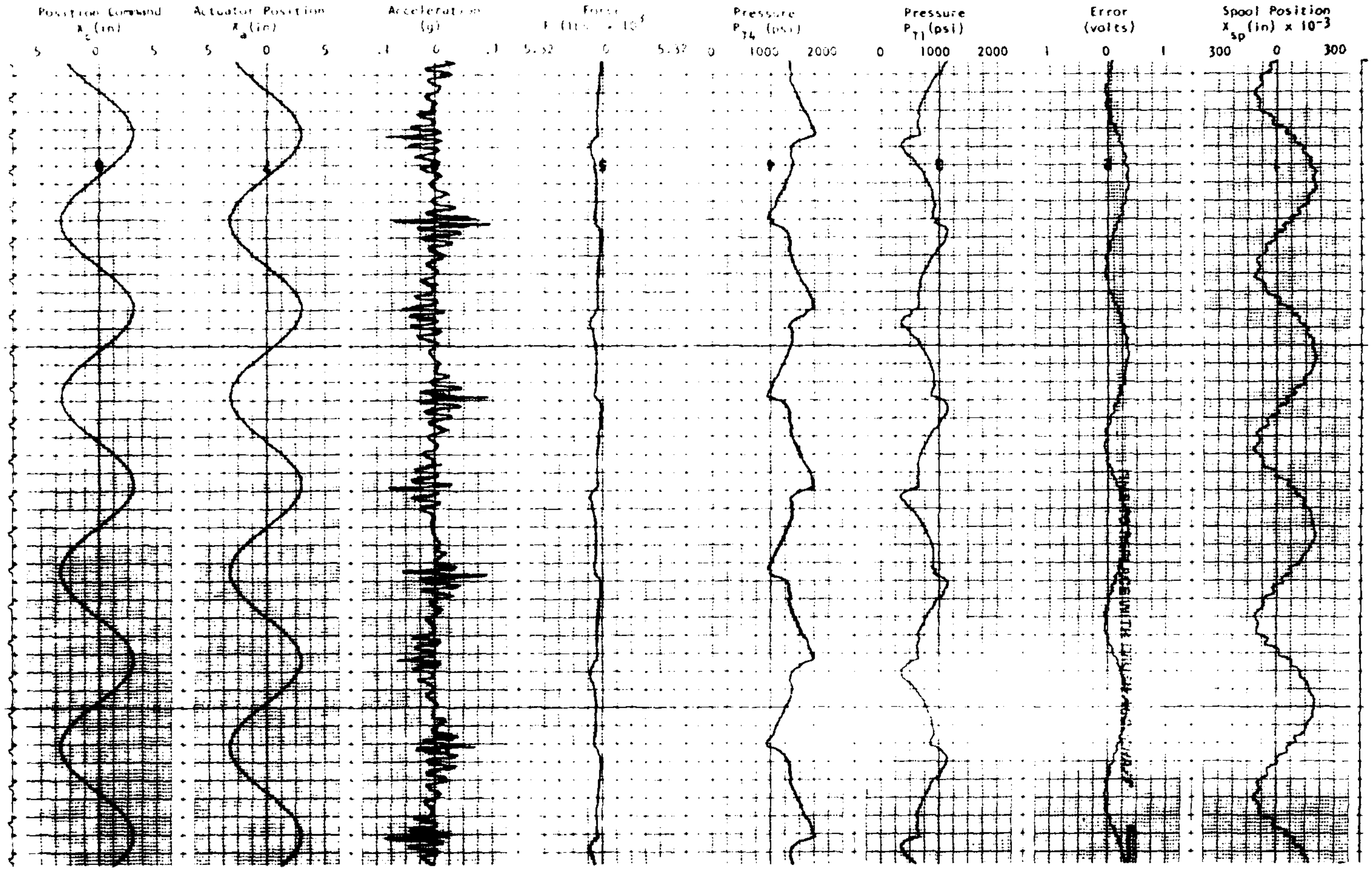
FIGURE: C-2



FREQUENCY: 0.10 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: EQUAL

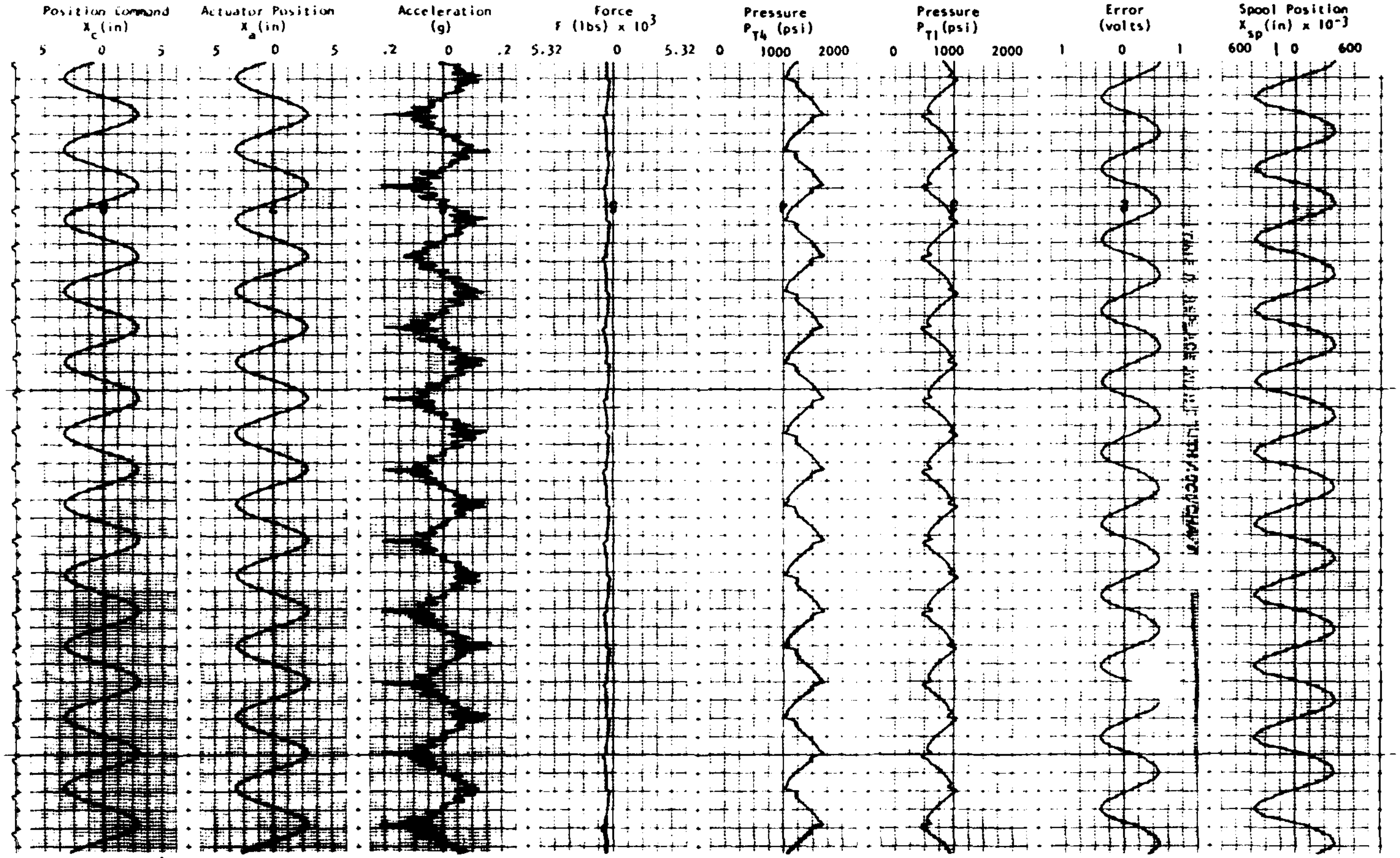
FIGURE: C-3



FREQUENCY: 0.20 **WAVEFORM:** SINE **LOAD:** BIASED

VALVE GAIN: HIGH **CYLINDER AREA:** EQUAL

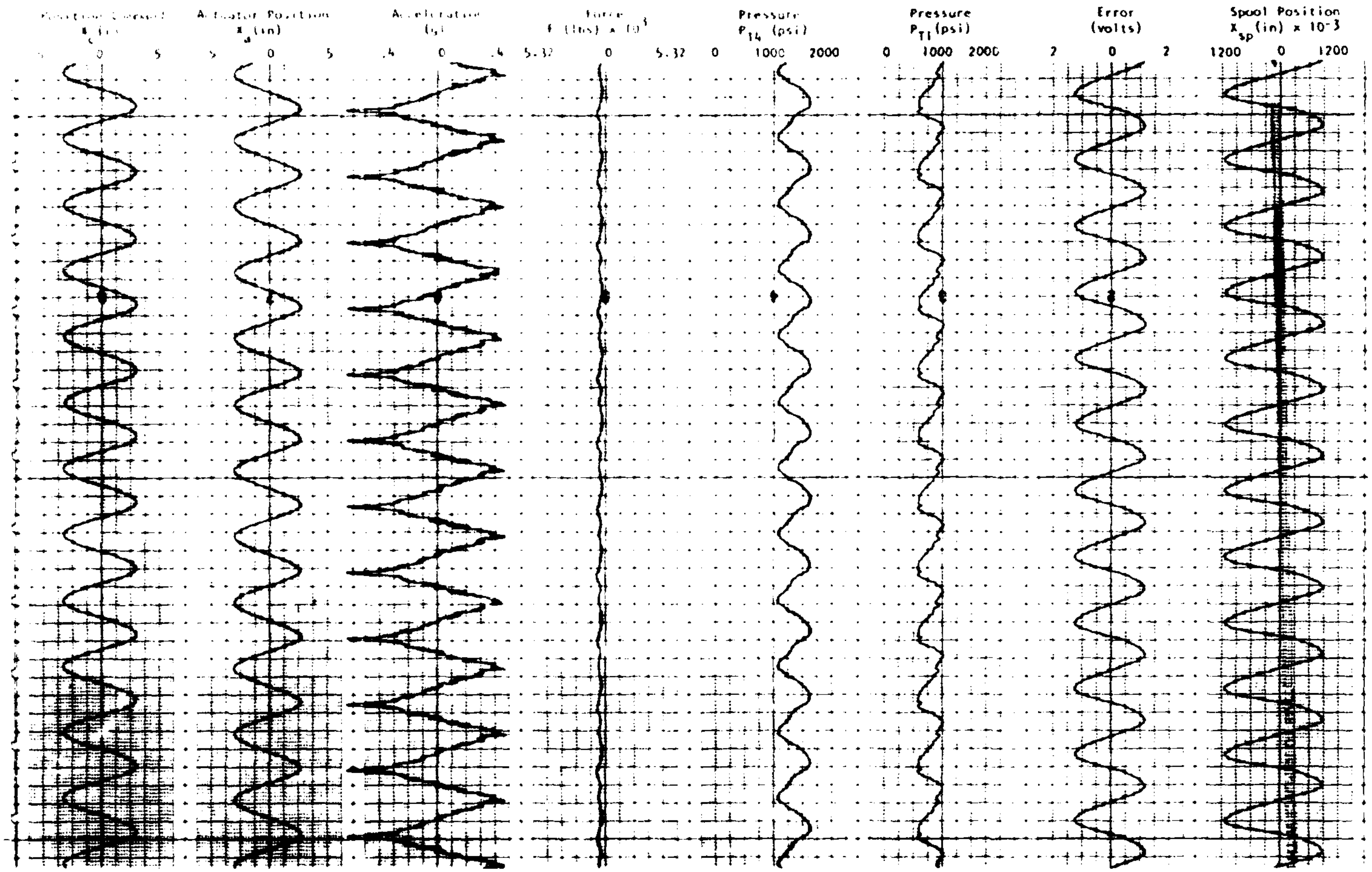
FIGURE: C - 4



FREQUENCY: 0.50 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: EQUAL

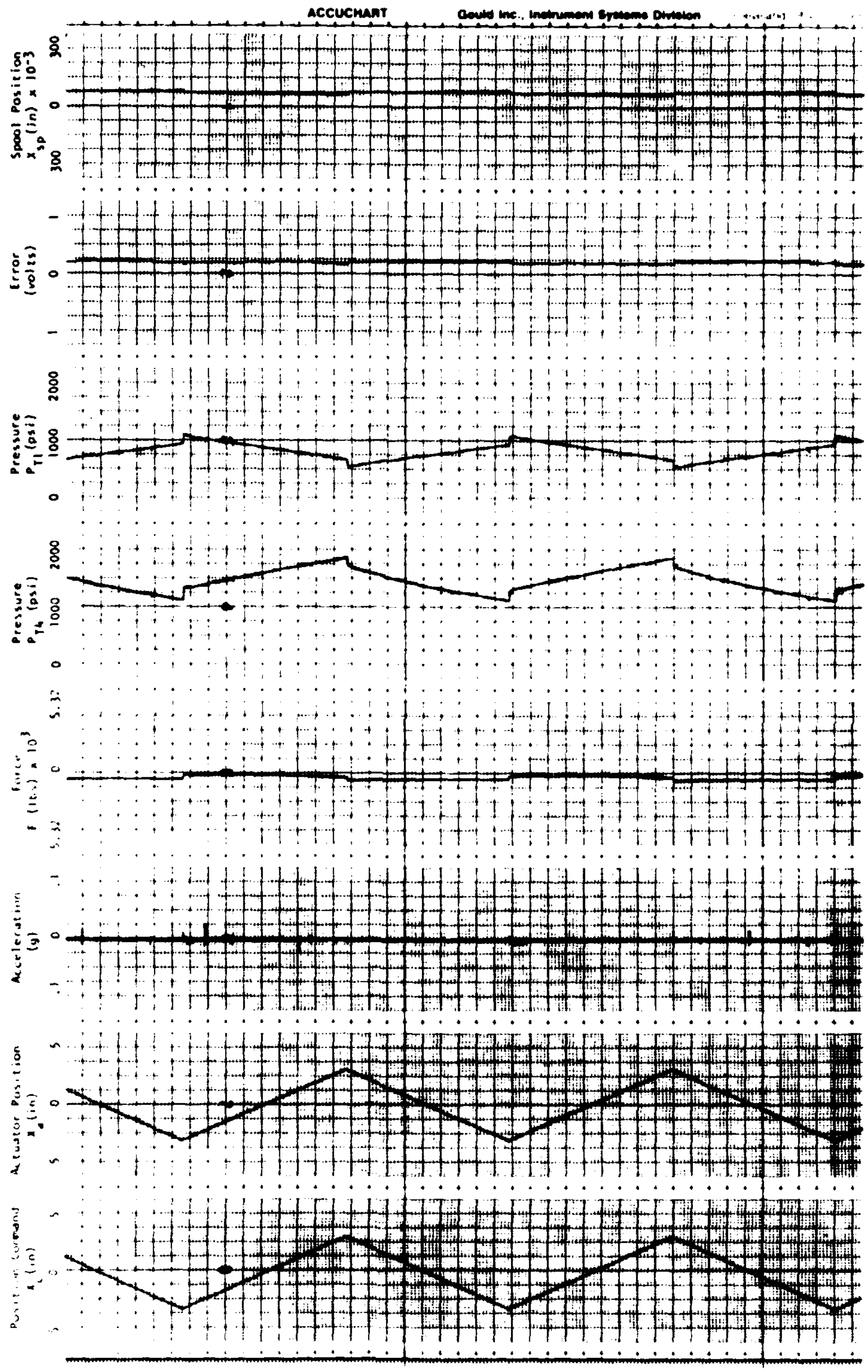
FIGURE: C-5



FREQUENCY: 1.00 **WAVEFORM:** SINE **LOAD:** BIASED

VALVE GAIN: HIGH **CYLINDER AREA:** EQUAL

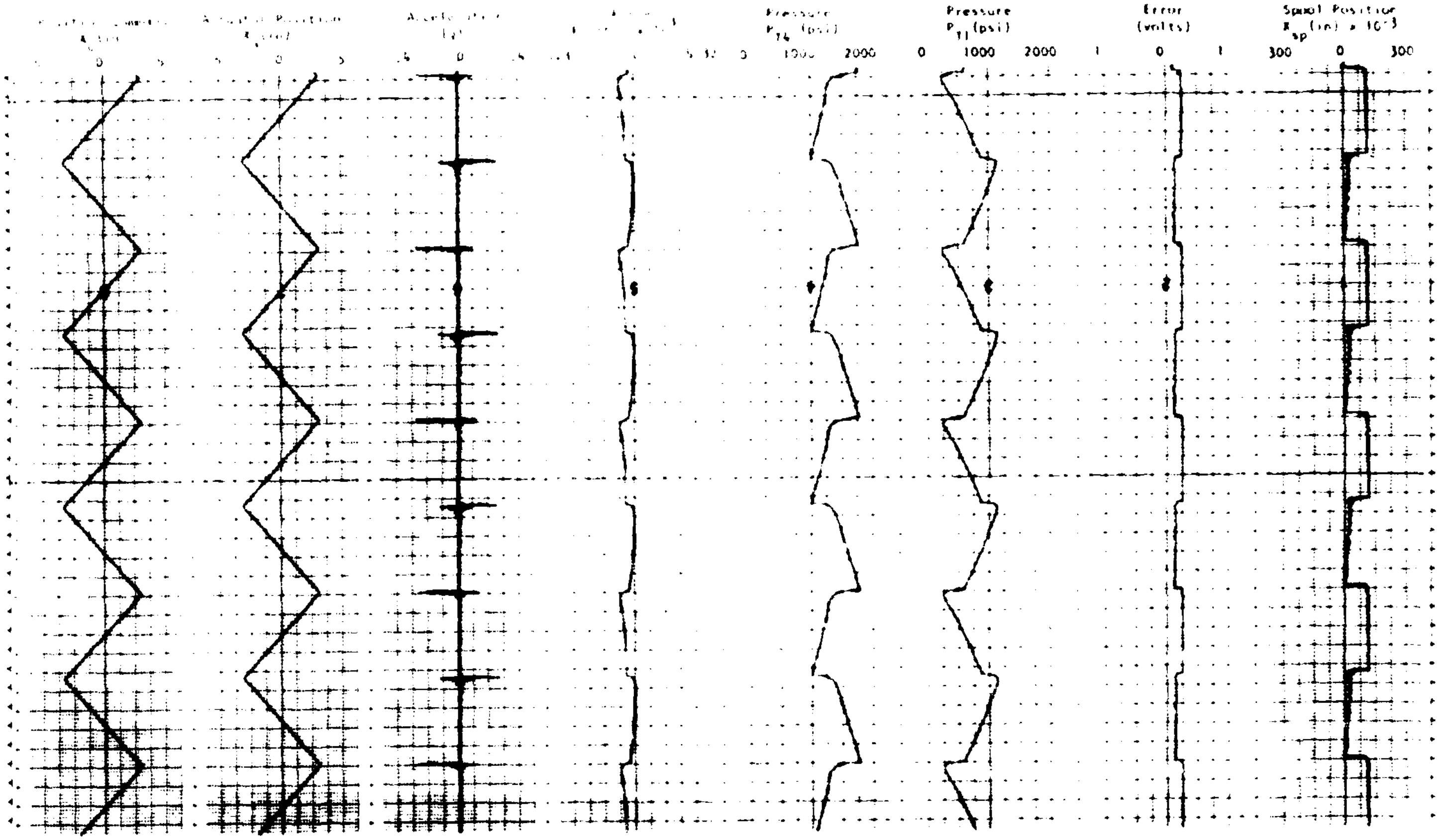
FIGURE: C-5



FREQUENCY: 0.01 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: EQUAL

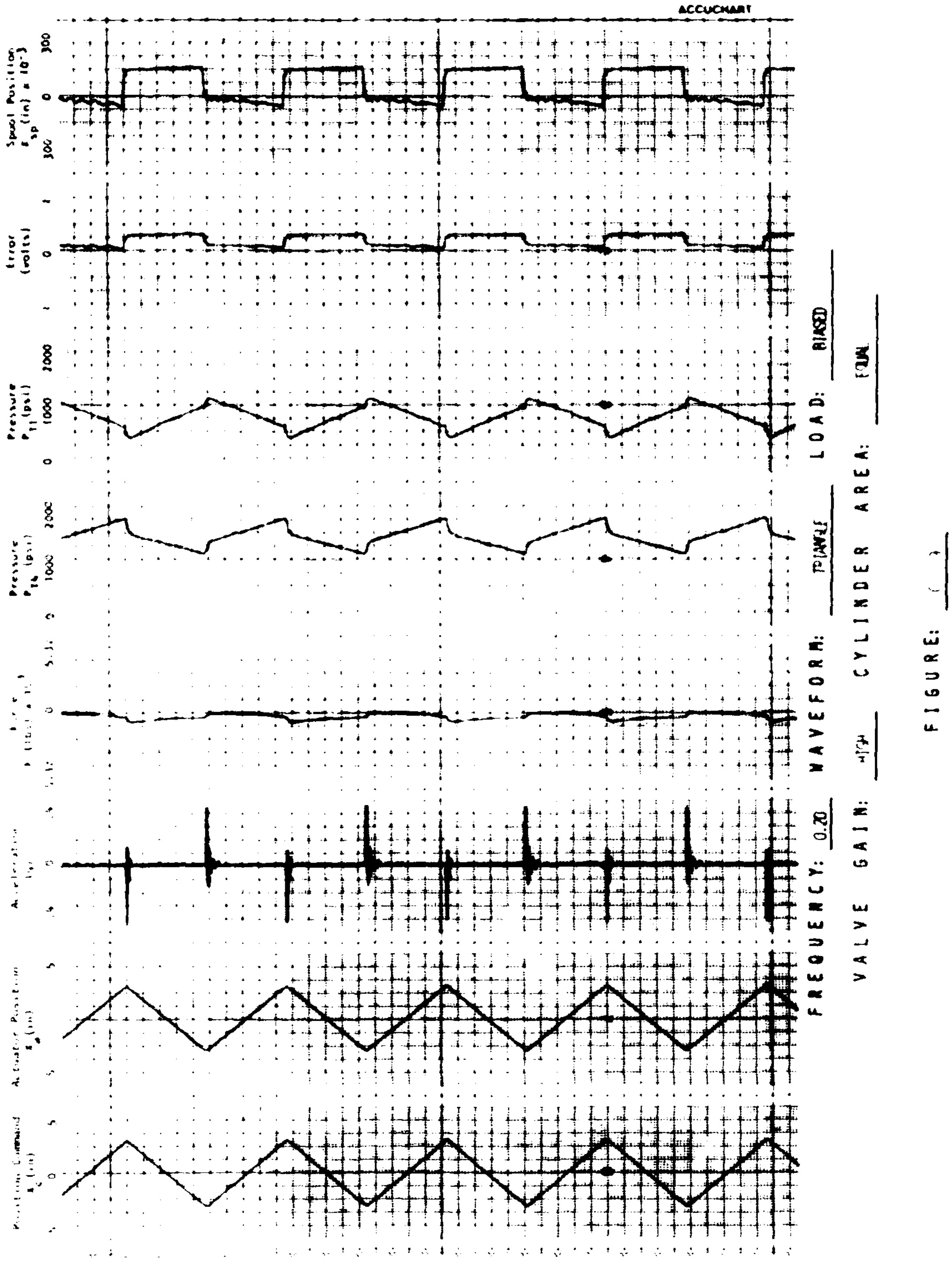
FIGURE: C-7

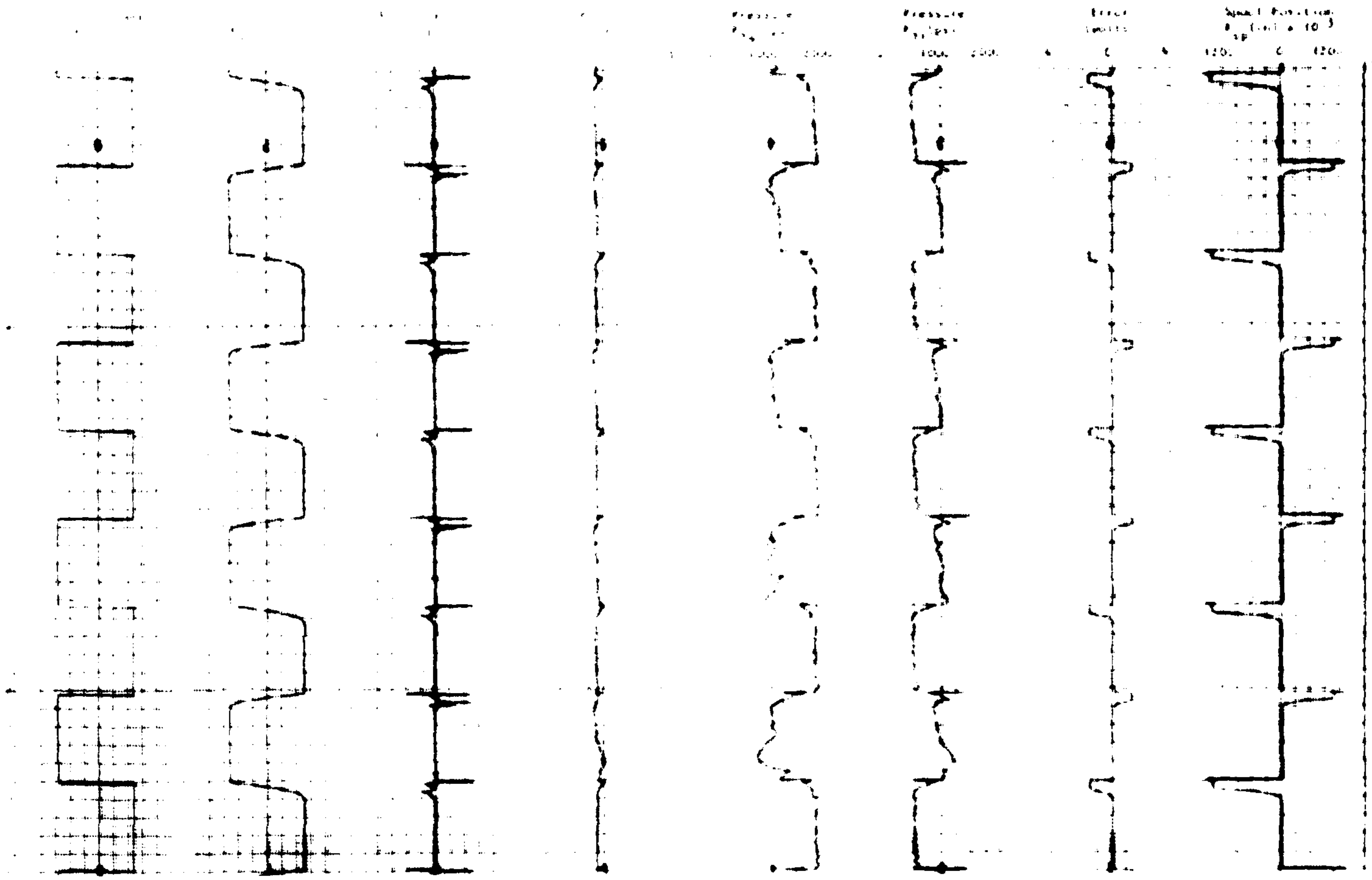


FREQUENCY: 0.10 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: HIGH CYLINDER AREA: EQUAL

FIGURE: C-8

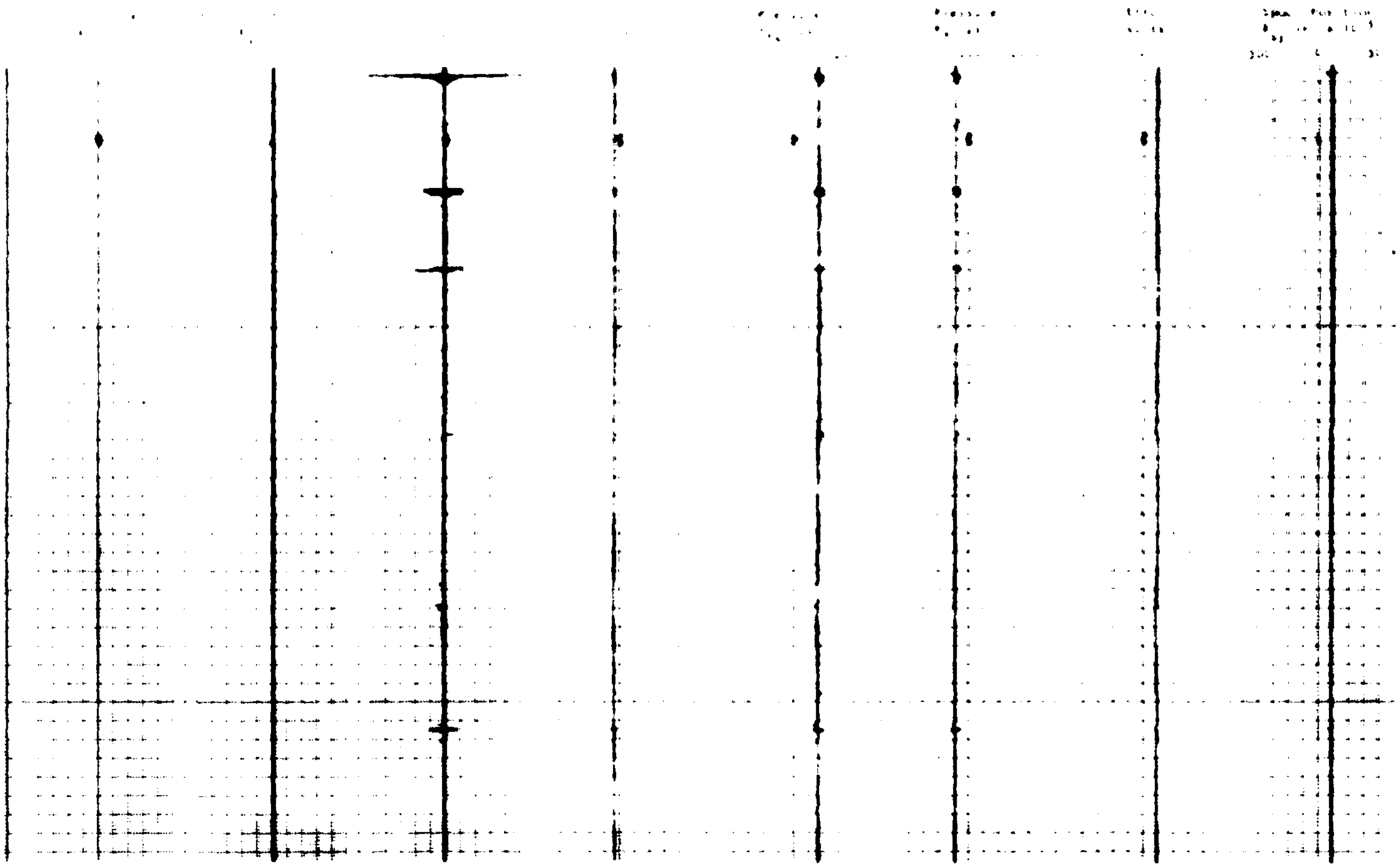

ACCUCHART
Spool Position
Error (volts)
Pressure
Pressure
Acceleration
Actuator Position
Position Command
FREQUENCY: 0.20
WAVEFORM: TRIANGLE
LOAD: BIASED
VALVE GAIN: HIGH
CYLINDER AREA: EQUAL
FIGURE:



FREQUENCY: ___ WAVEFORM ___ LOAD: ___

VALVE GAIN: ___ CYLINDER AREA ___

FIGURE: ___



FREQUENCY: [illegible] WAVEFORM: [illegible] LOAD: BIASED

VALVE GAIN: [illegible] CYLINDER AREA: [illegible]

FIGURE: [illegible]-11

APPENDIX

D

Small Scale System Tests,
Franklin Low Gain Valve with Equal Area Cylinders

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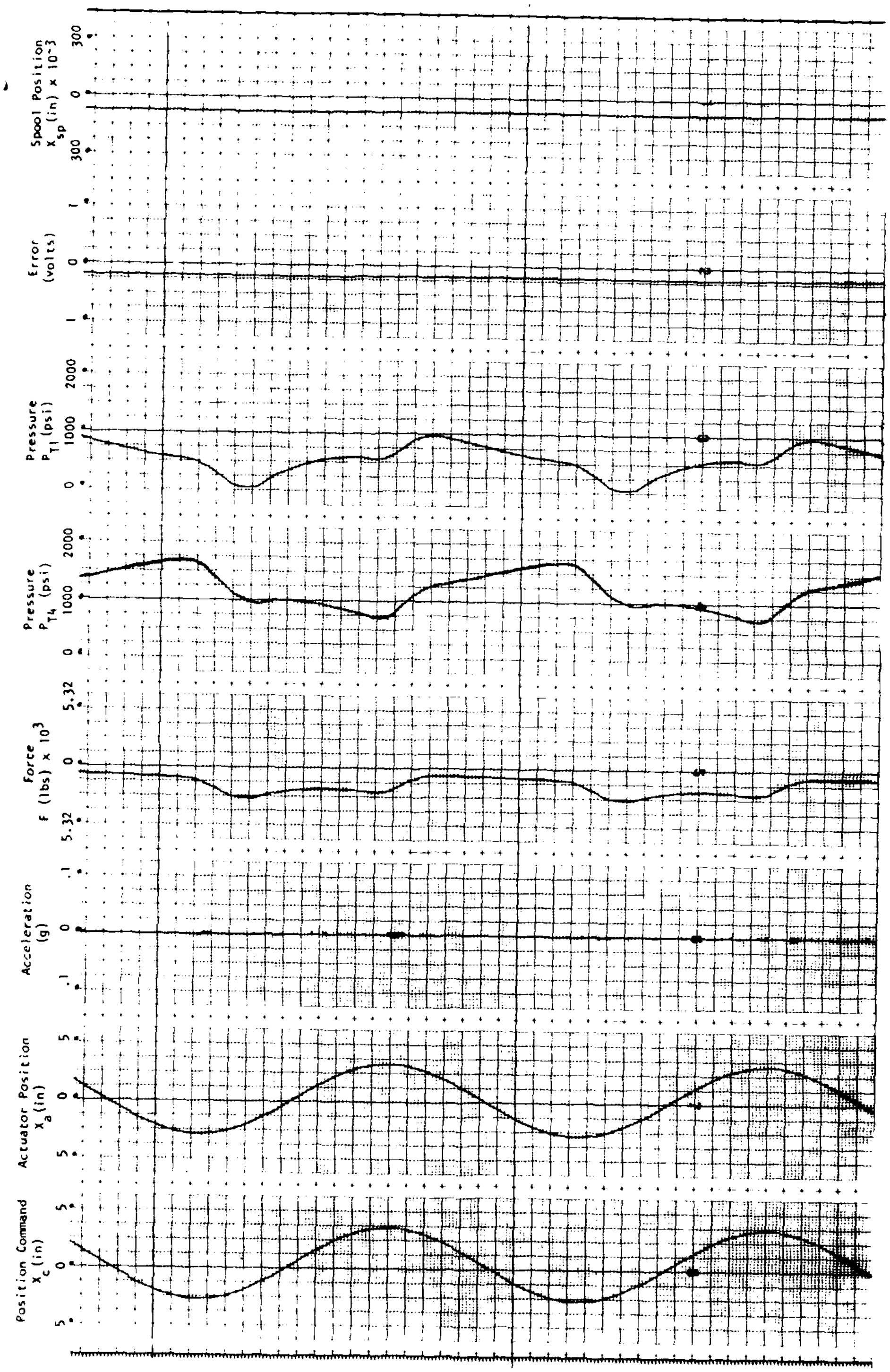
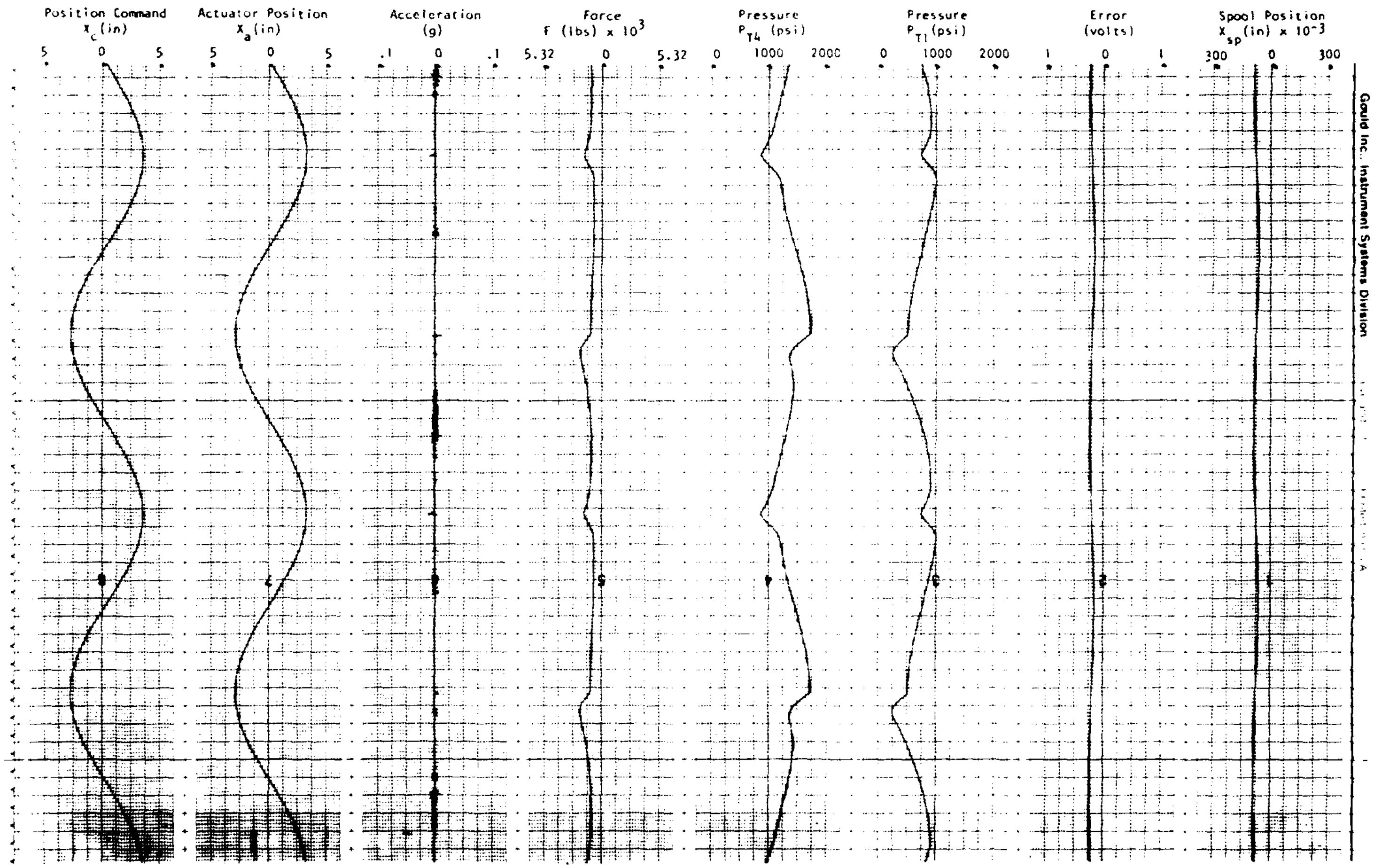


FREQUENCY: 0.01 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

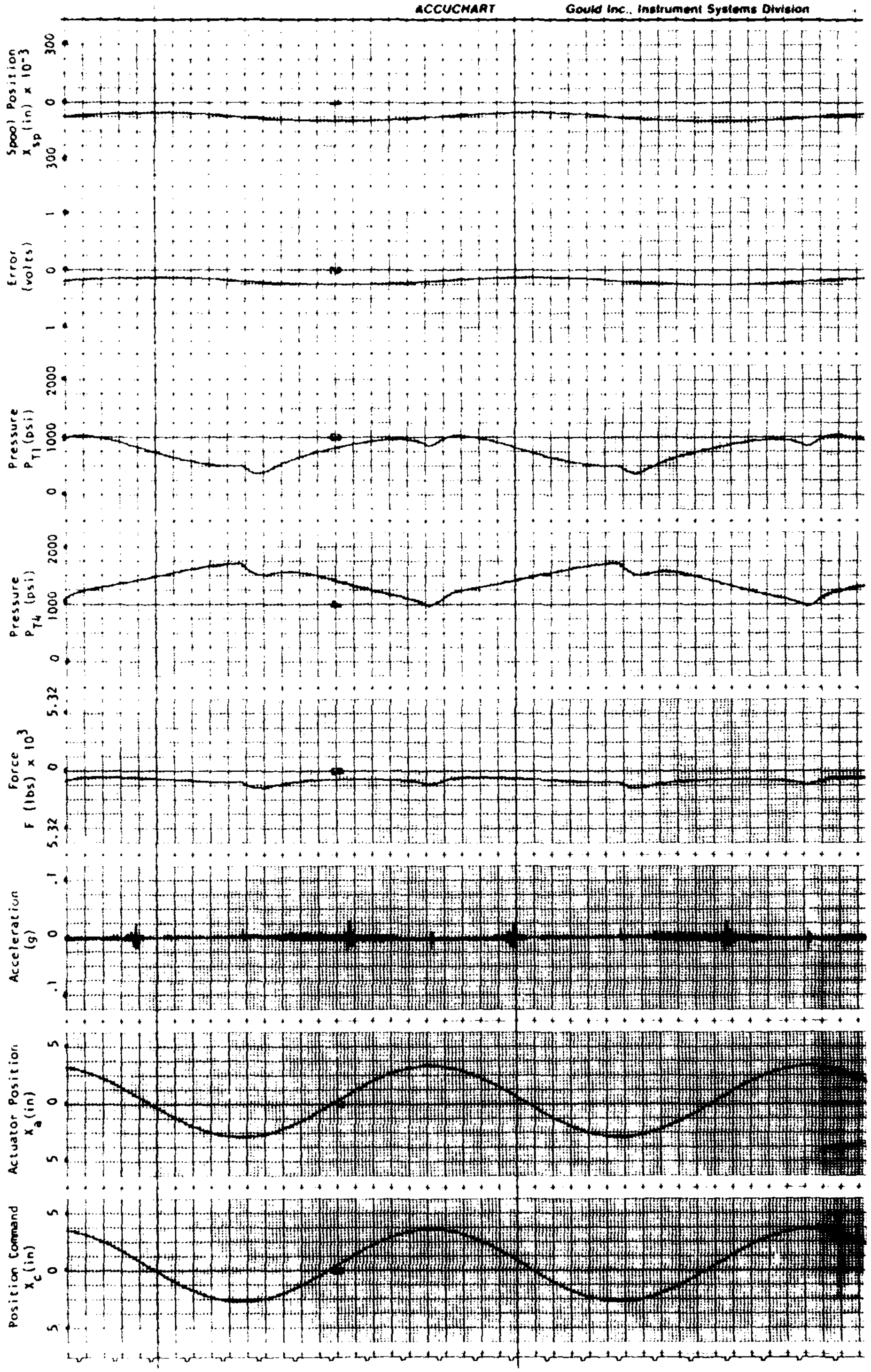
FIGURE: D-1



FREQUENCY: 0.05 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

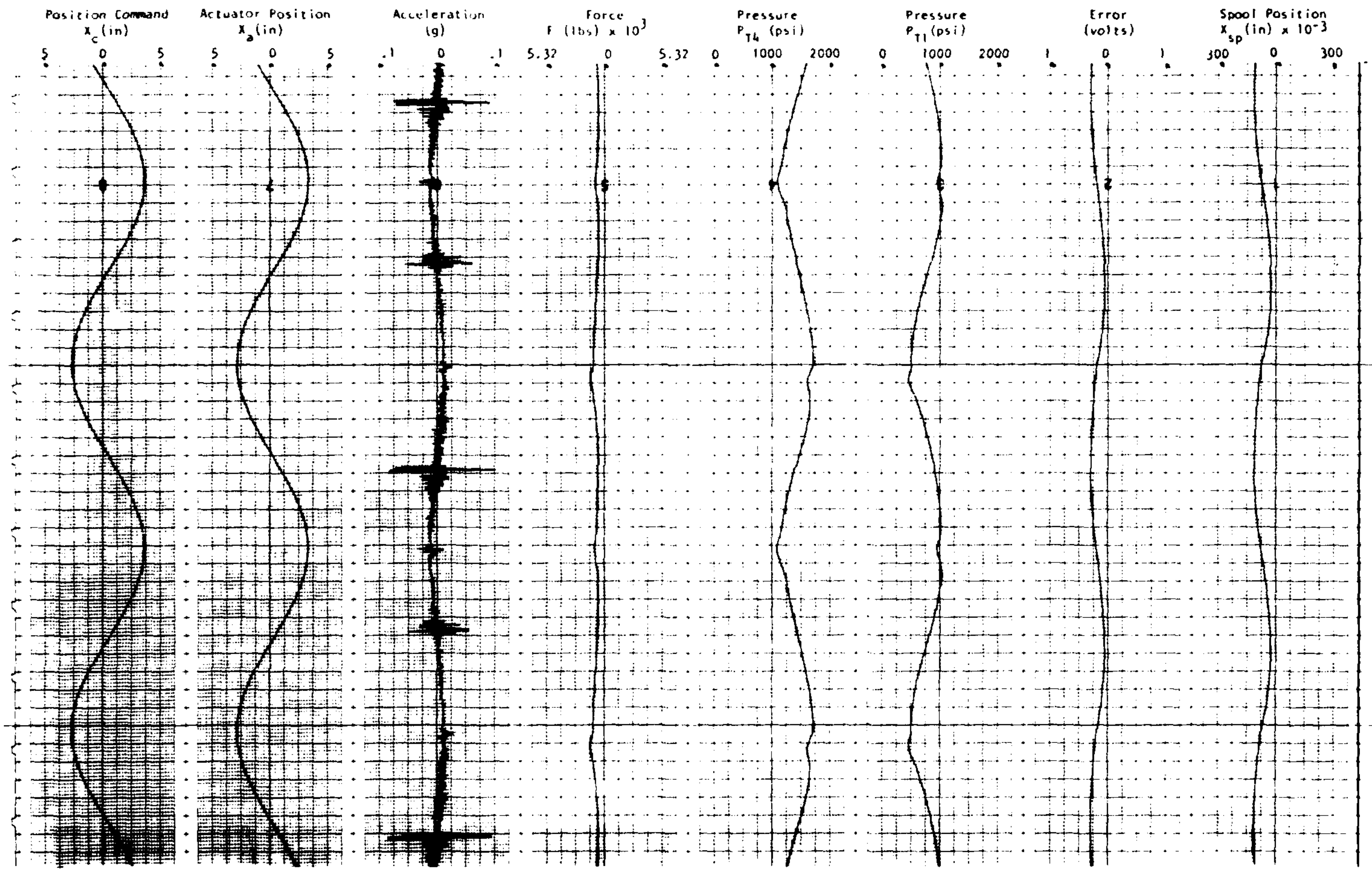
FIGURE: II-2



FREQUENCY: 0.10 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

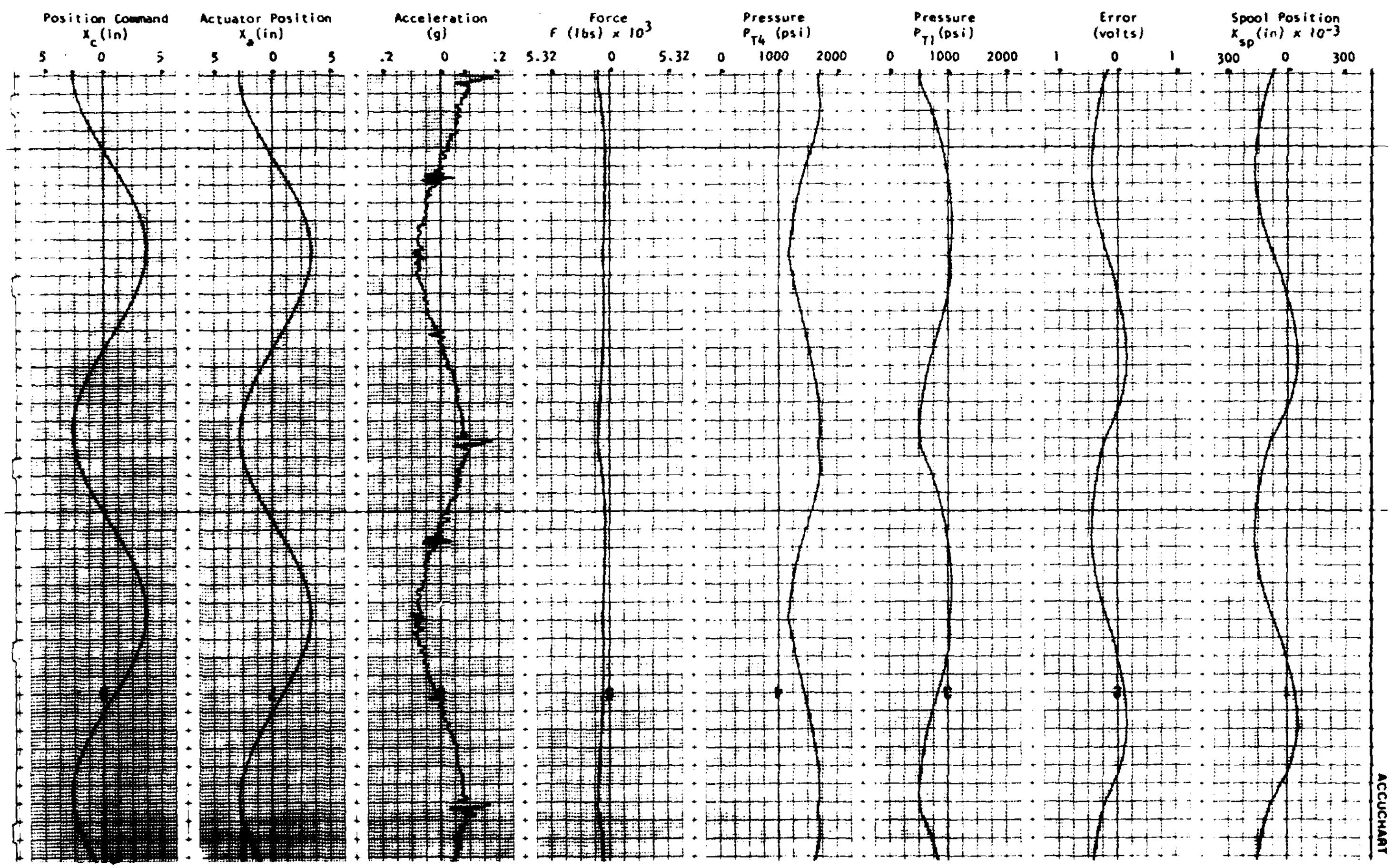
FIGURE: D-3



FREQUENCY: 020 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

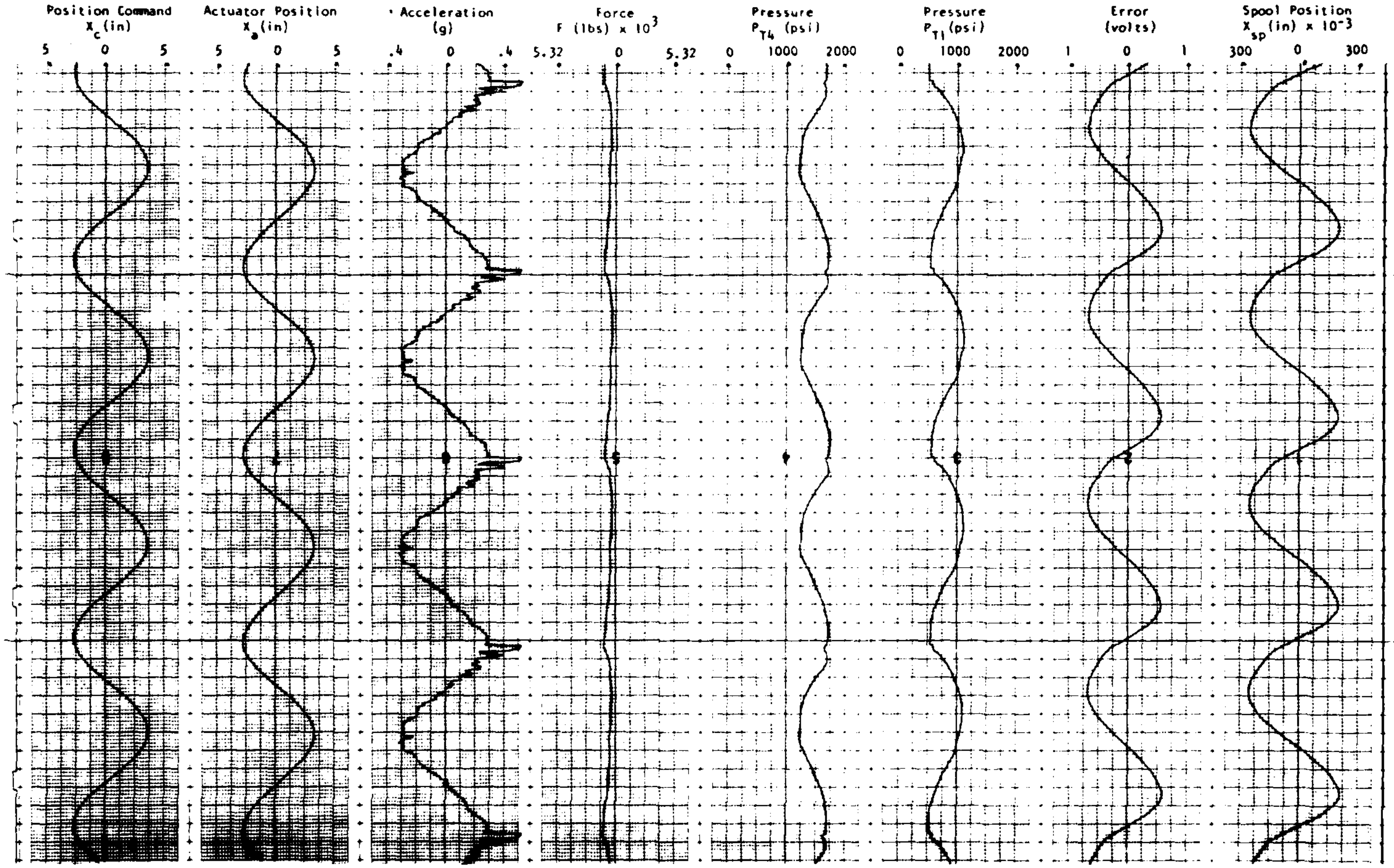
FIGURE: D-4



FREQUENCY: 0.50 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

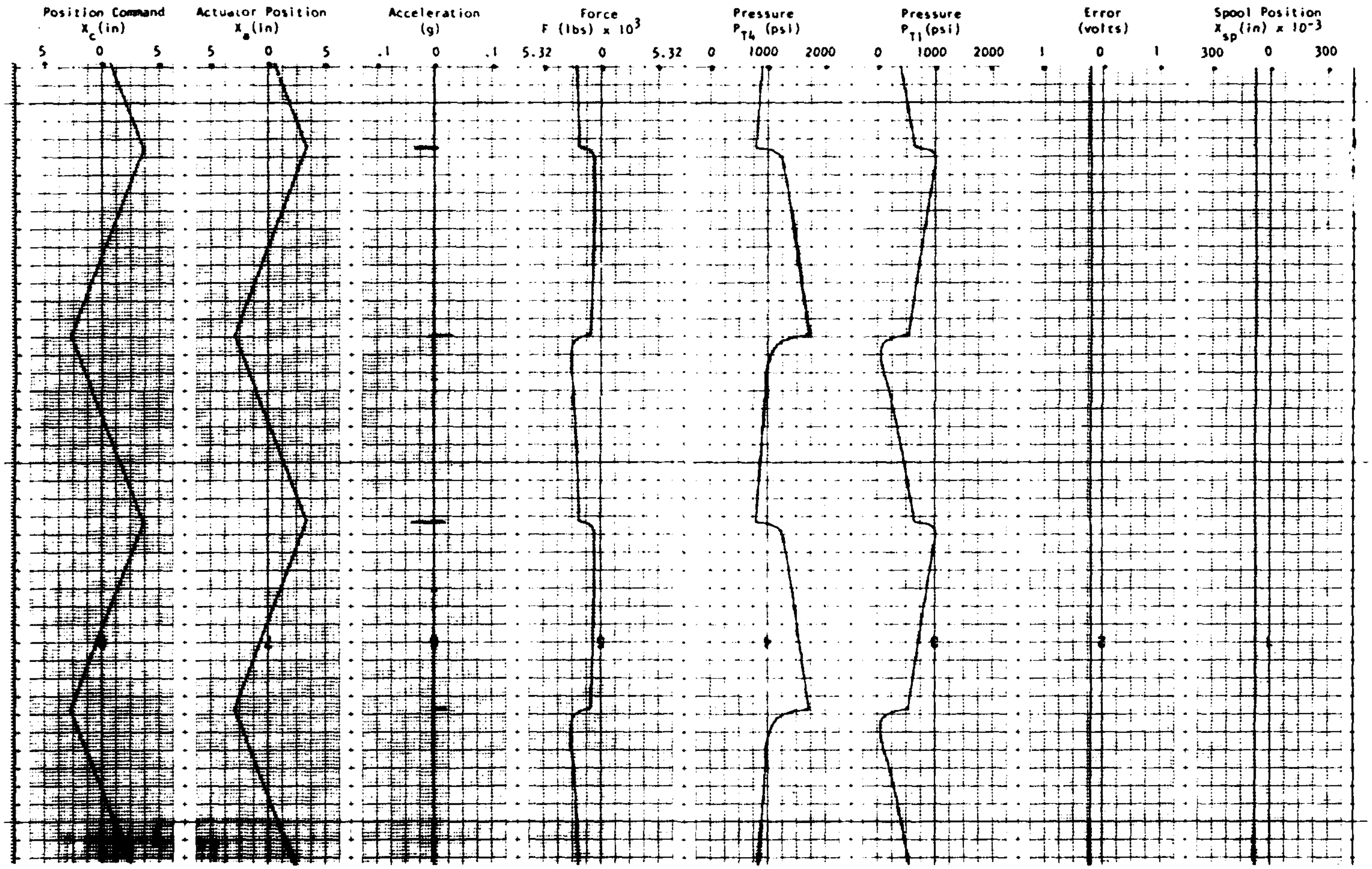
FIGURE: D-5



FREQUENCY: 1.00 WAVEFORM: SINE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

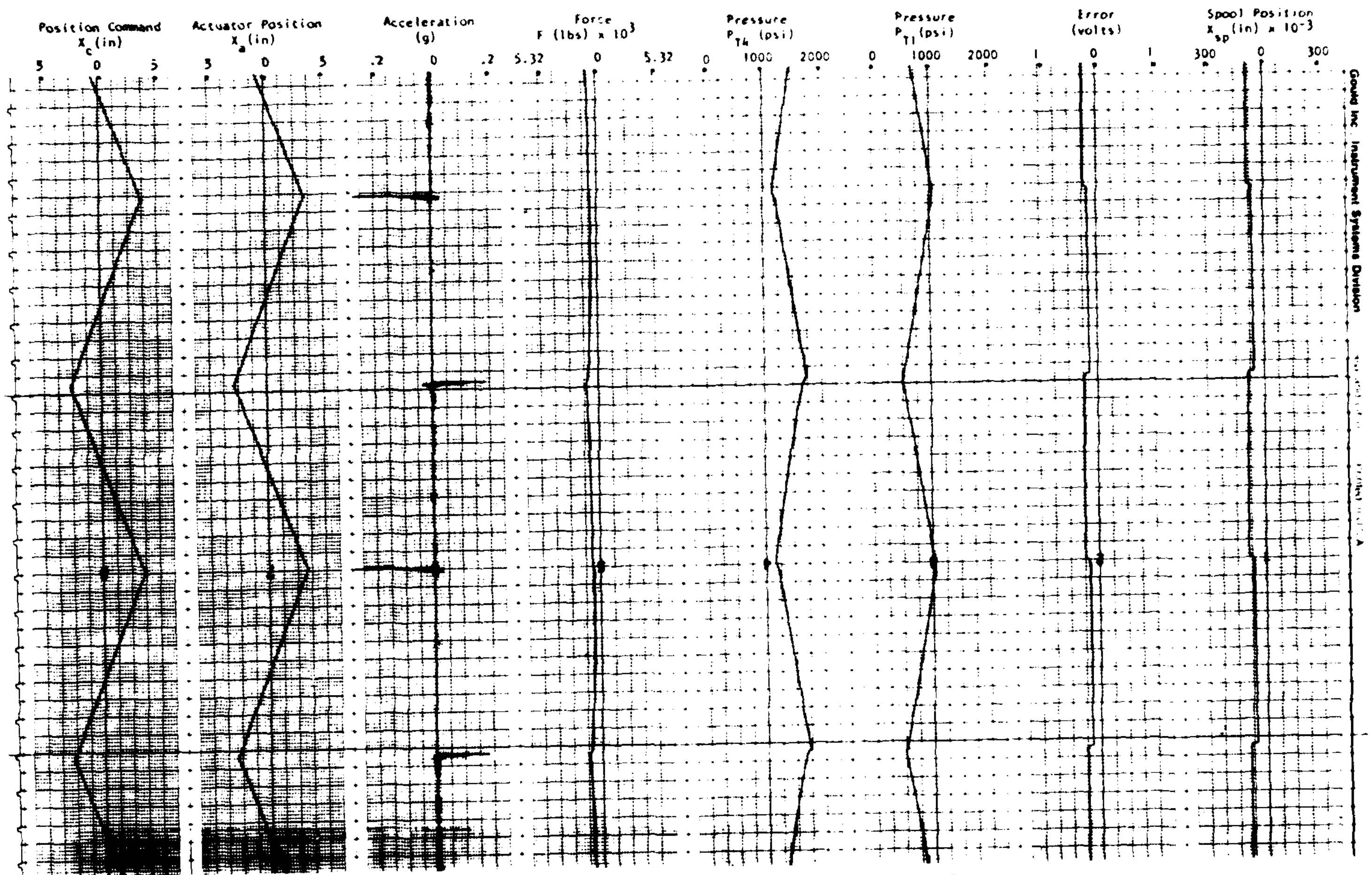
FIGURE: D-6



FREQUENCY: 0.01 WAVEFORM: TRIANGLE LOAD: BIAS

VALVE GAIN: LOW CYLINDER AREA: EQUAL

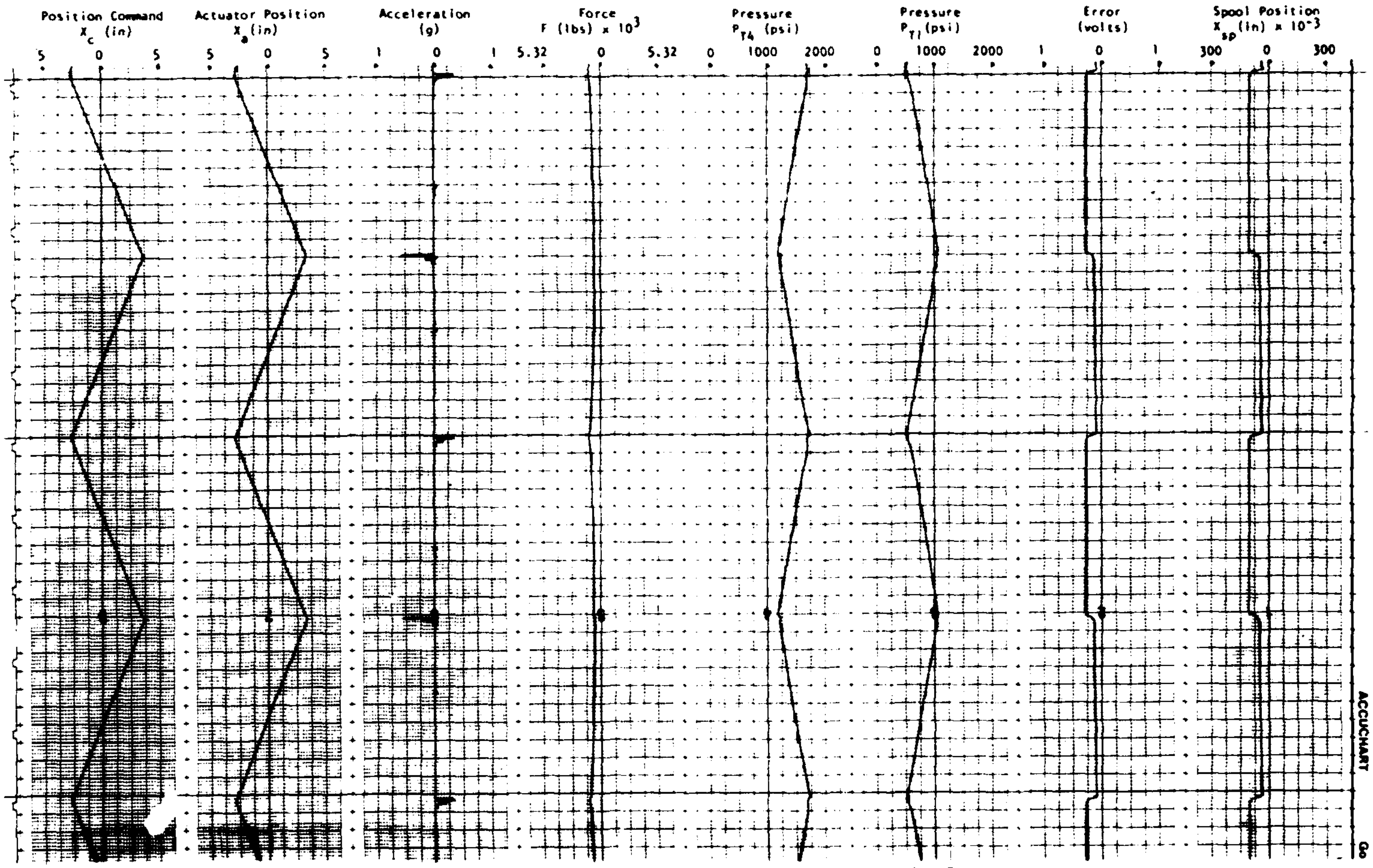
FIGURE: D-7



FREQUENCY: 0.10 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

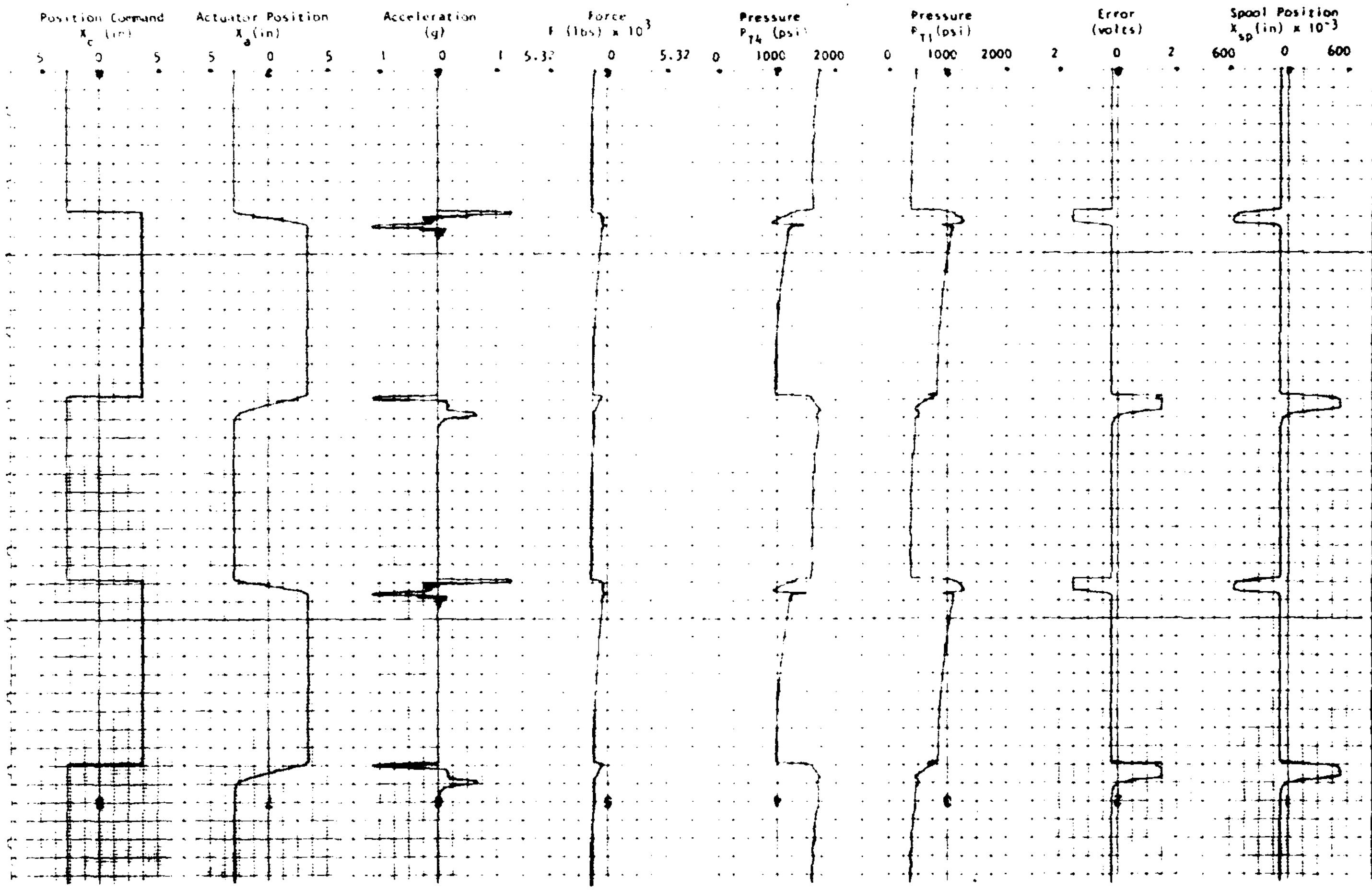
FIGURE: D-9



FREQUENCY: 0.20 WAVEFORM: TRIANGLE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

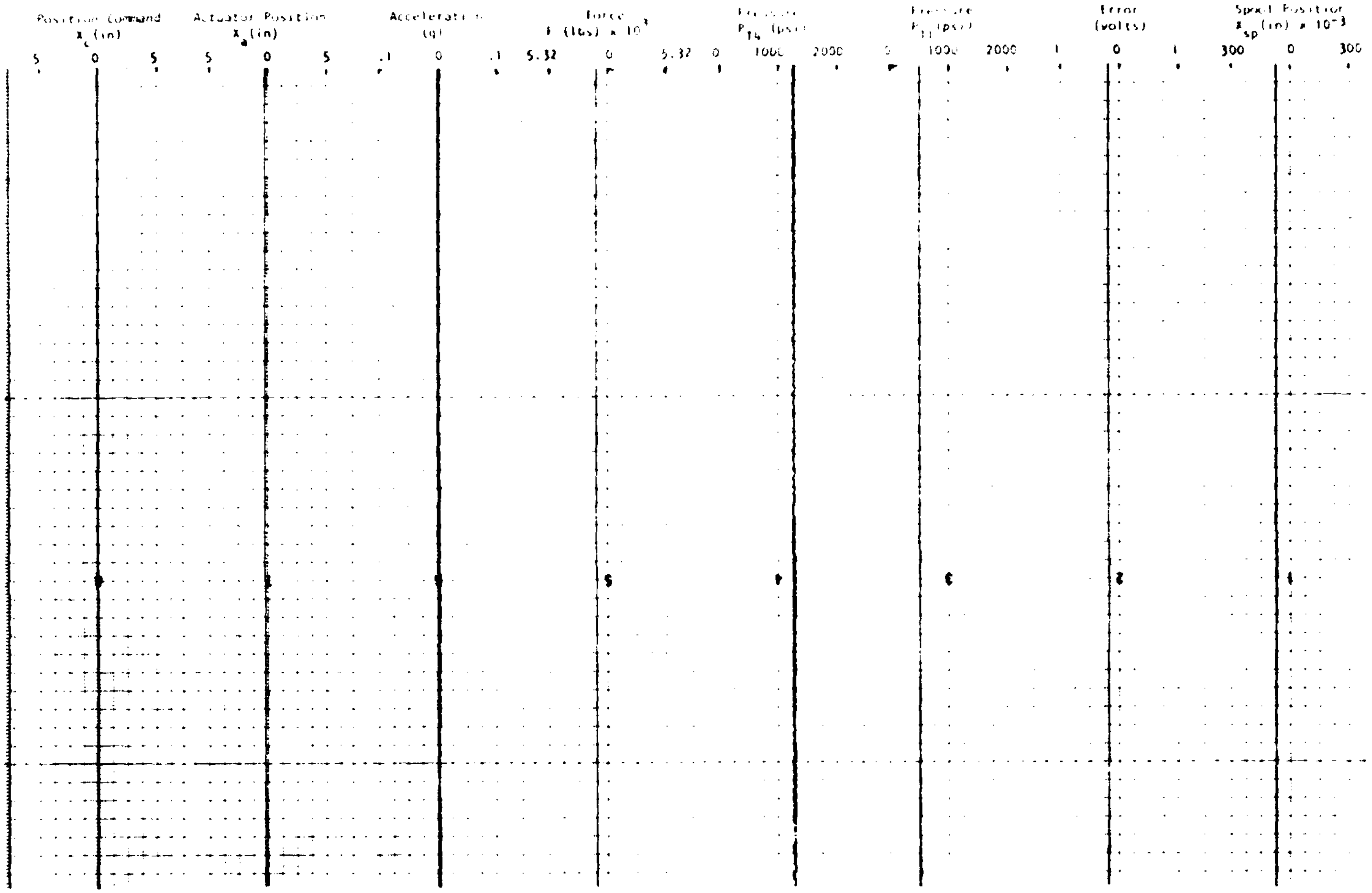
FIGURE: D-10



FREQUENCY: 0.20 WAVEFORM: SQUARE LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

FIGURE: B-10



FREQUENCY: 0.00 WAVEFORM: ZERO LOAD: BIASED

VALVE GAIN: LOW CYLINDER AREA: EQUAL

FIGURE: [illegible]

APPENDIX

E

Full Scale System Tests,
Commercial High Gain Valve

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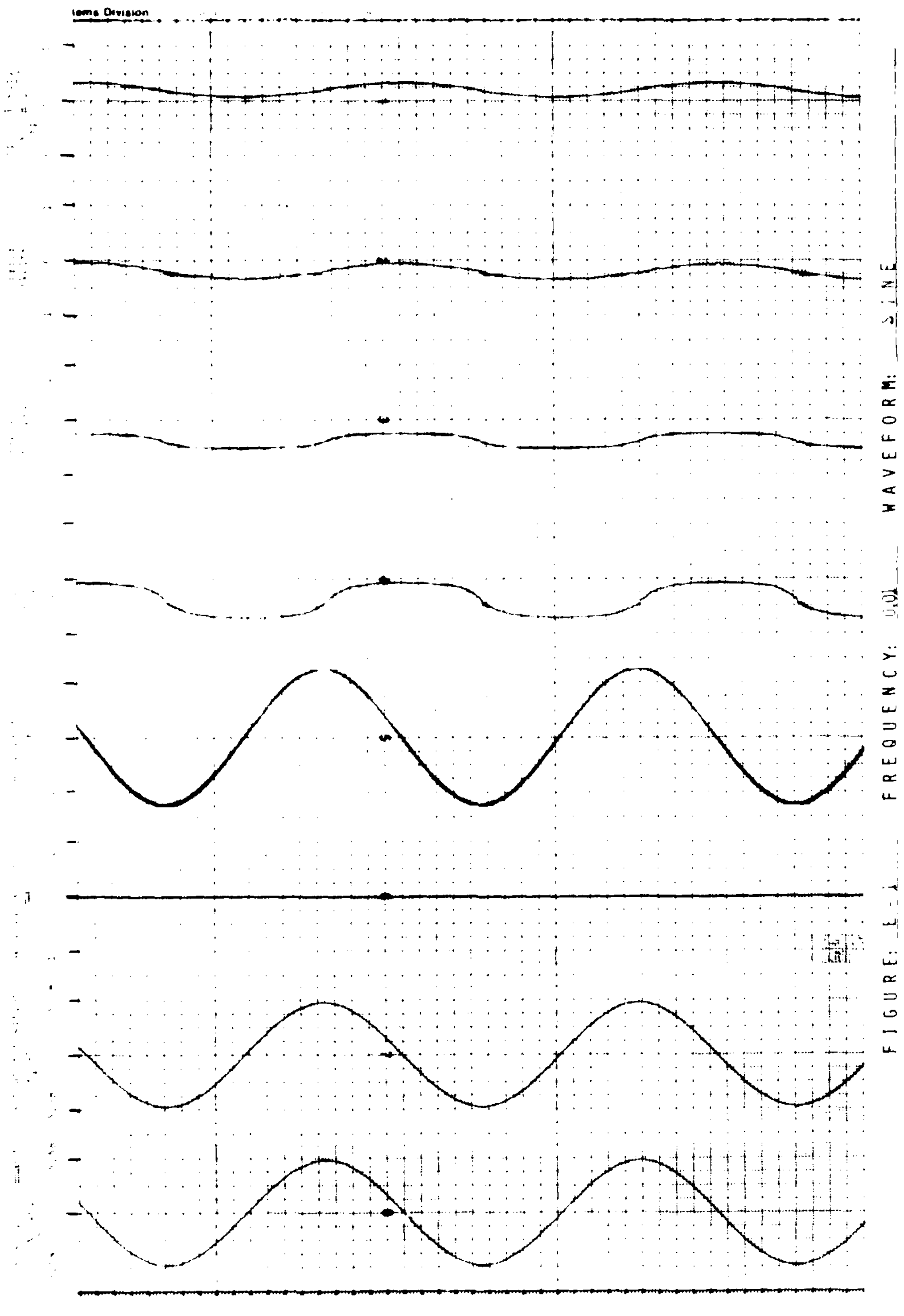

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AD-A116 106 FRANKLIN RESEARCH CENTER PHILADELPHIA PA F/G 13/7
INVESTIGATION OF 'HYDRAULIC BUMP' IN SIMULATOR ELECTROHYDRAULIC--ETC(U)
AUG 81 C A BELSTERLING, K S FERTNER, J STONE F33657-80-C-0206
UNCLASSIFIED FRC-F-C5364 ASD-TR-81-5033 NL

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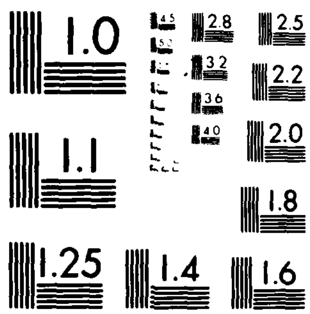

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS 1963-A

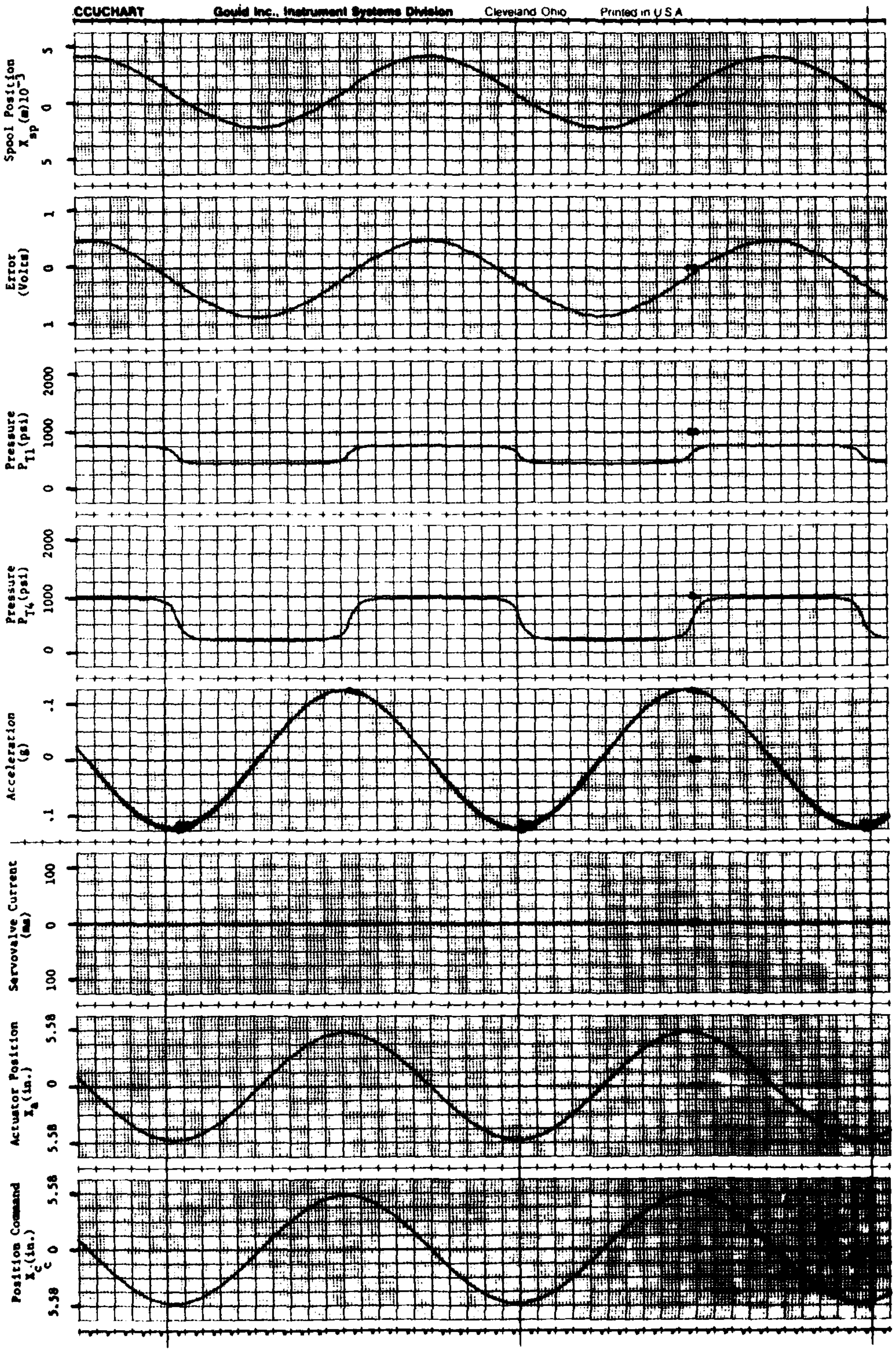

FIGURE: E-2 FREQUENCY: 0.05 WAVEFORM: SINE

VALVE GAIN: HIGH

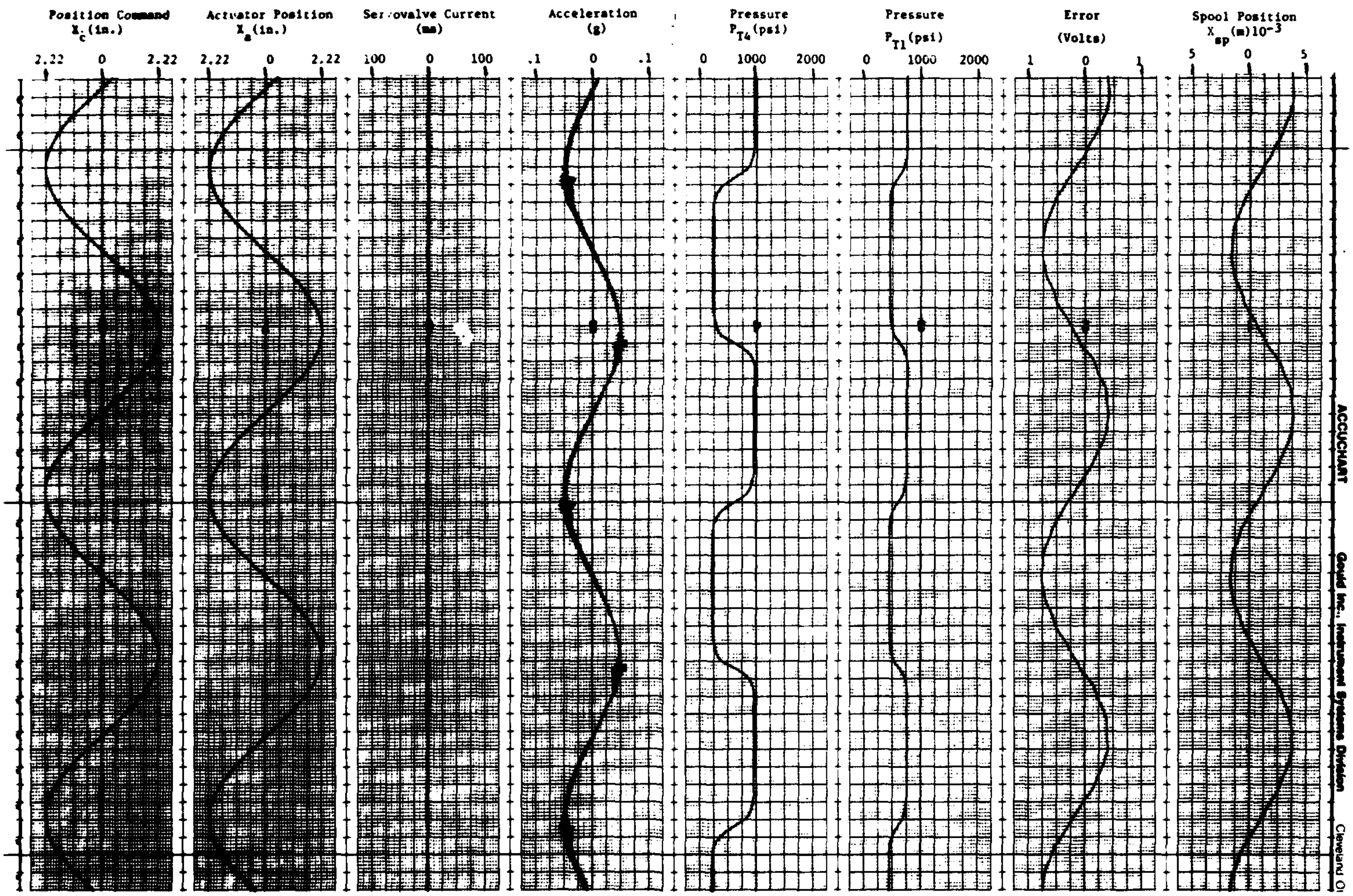

FIGURE: E-3 FREQUENCY: 0.10 WAVEFORM: SINE

VALVE GAIN: HIGH

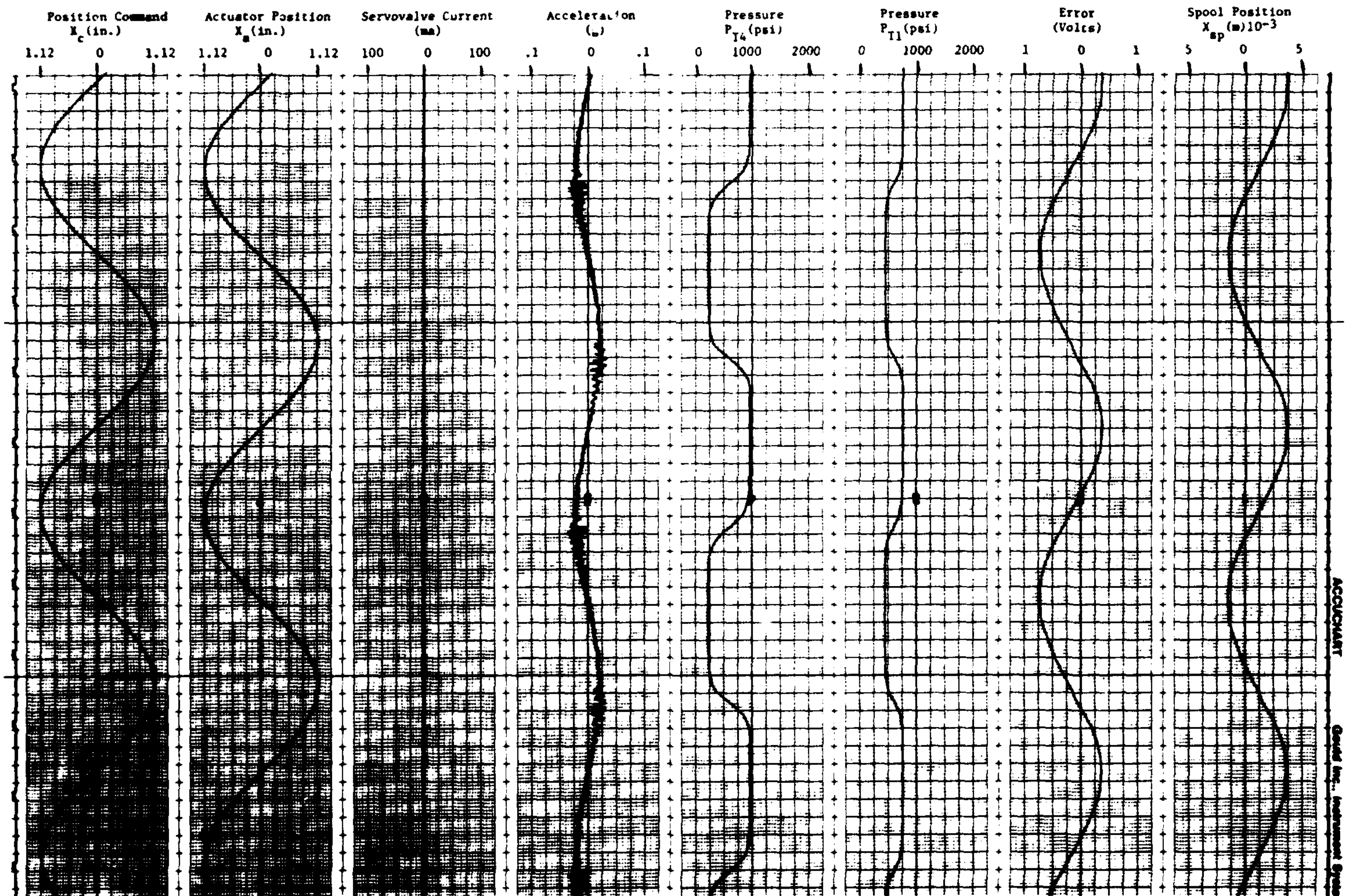

FIGURE: E-4 FREQUENCY: 0.20 WAVEFORM: SINE

VALVE GAIN: HIGH

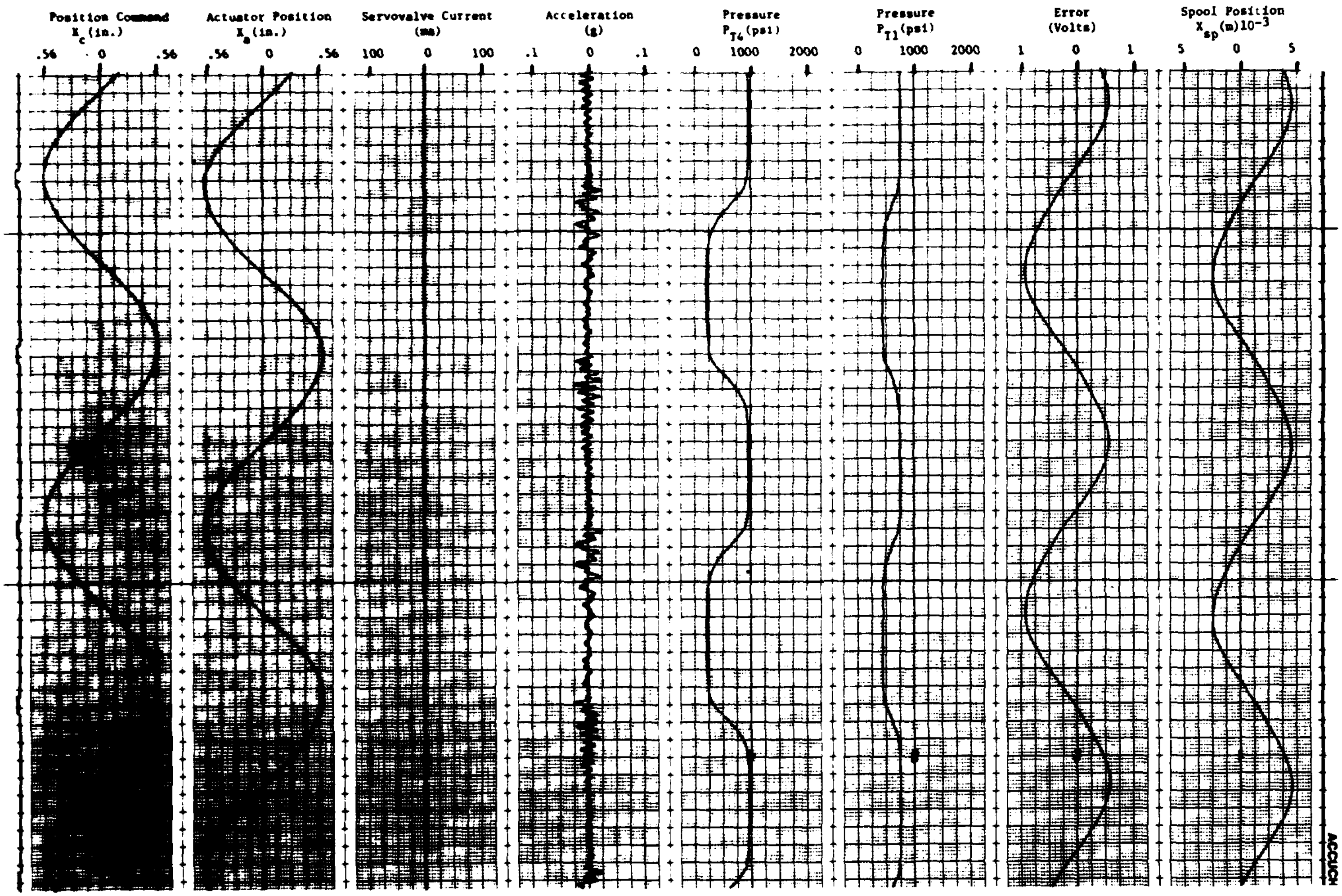

FIGURE: E-5 FREQUENCY: 0.50 WAVEFORM: SINE

VALVE GAIN: HIGH

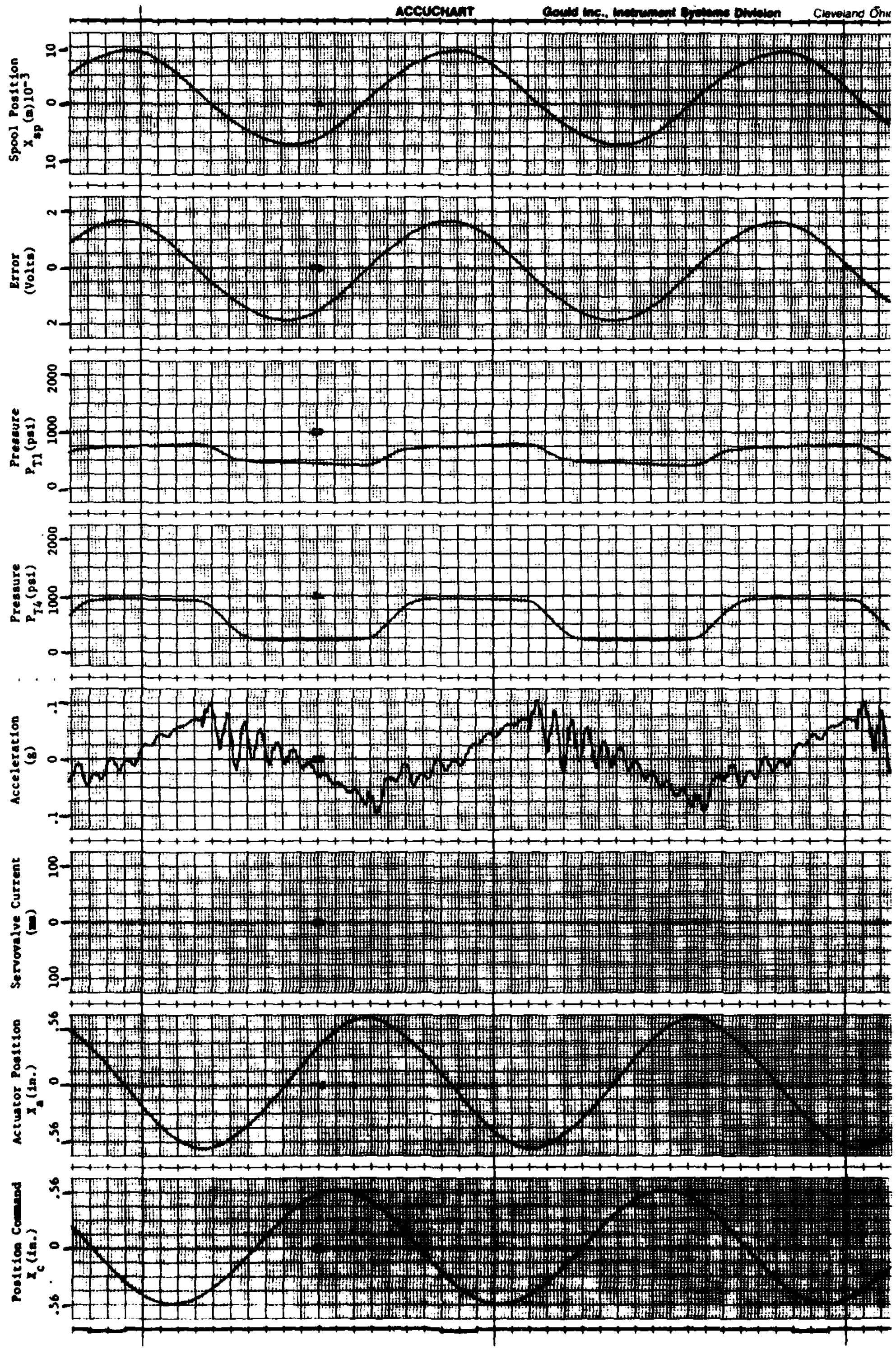

FIGURE: E-6 FREQUENCY: 1.00 WAVEFORM: SINE

VALVE GAIN: HIGH

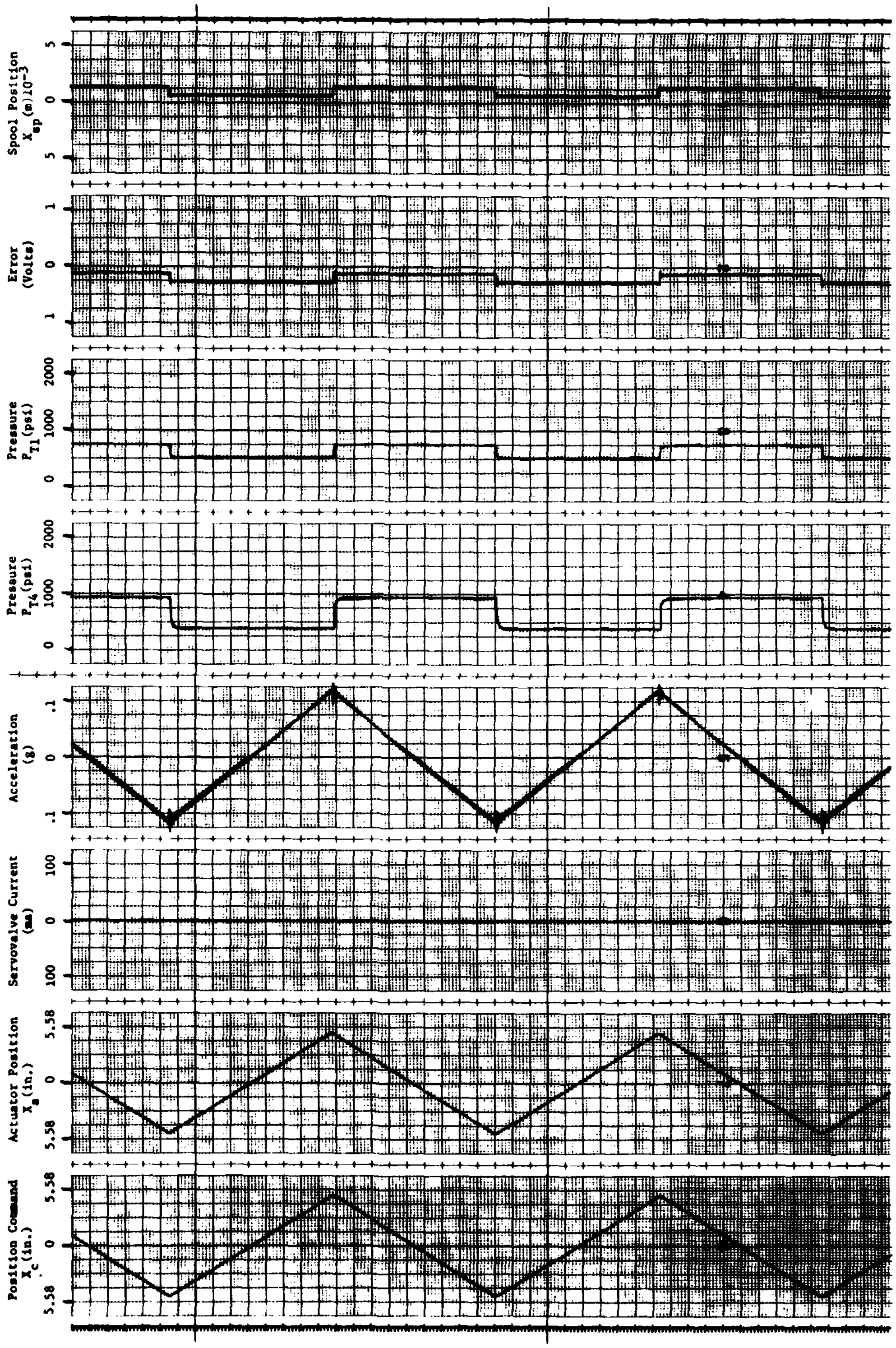

FIGURE: E-7 FREQUENCY: 0.01 WAVEFORM: TRIANGLE

VALVE GAIN: HIGH

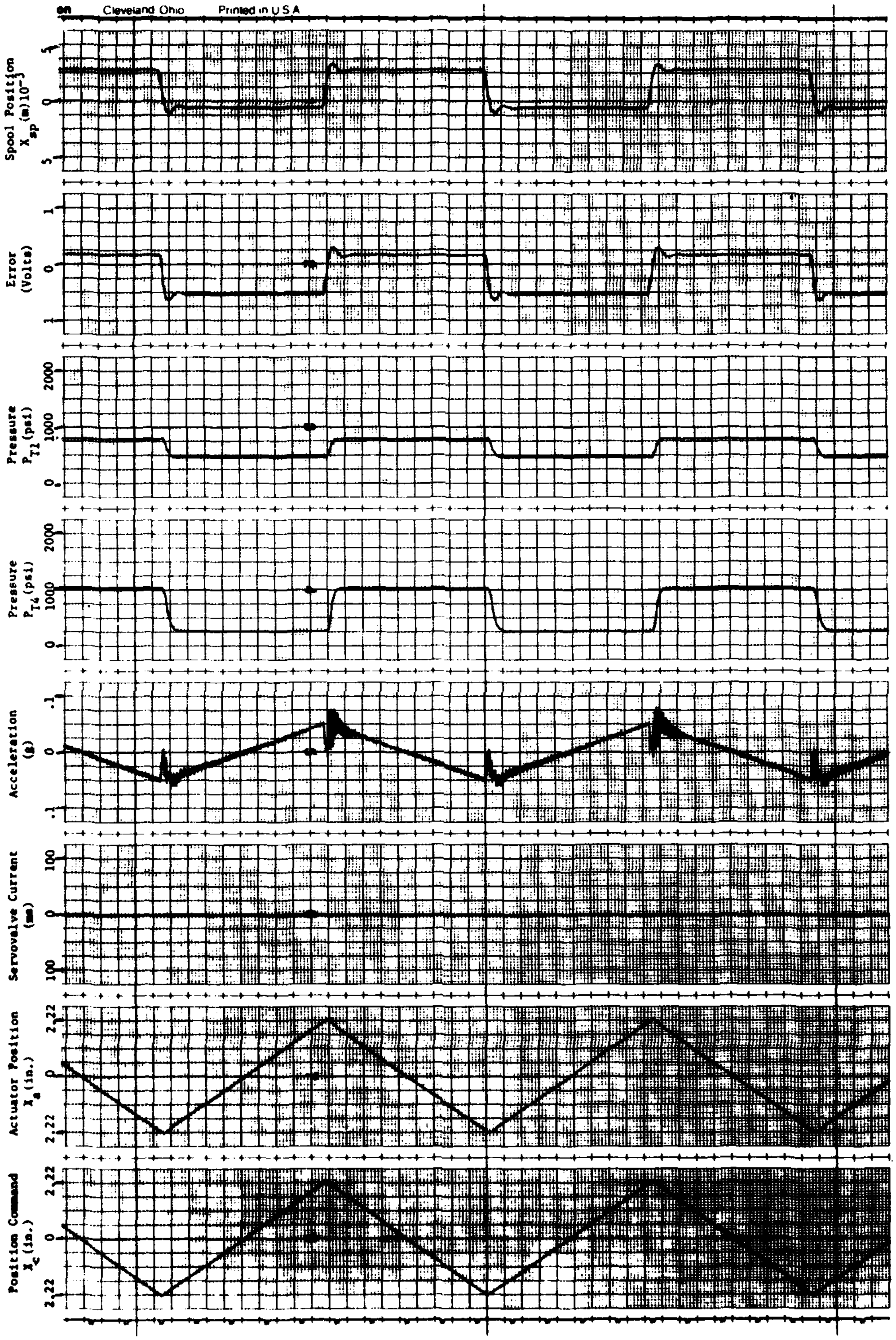

FIGURE: E-8 FREQUENCY: 0.10 WAVEFORM: TRIANGLE

VALVE GAIN: HIGH

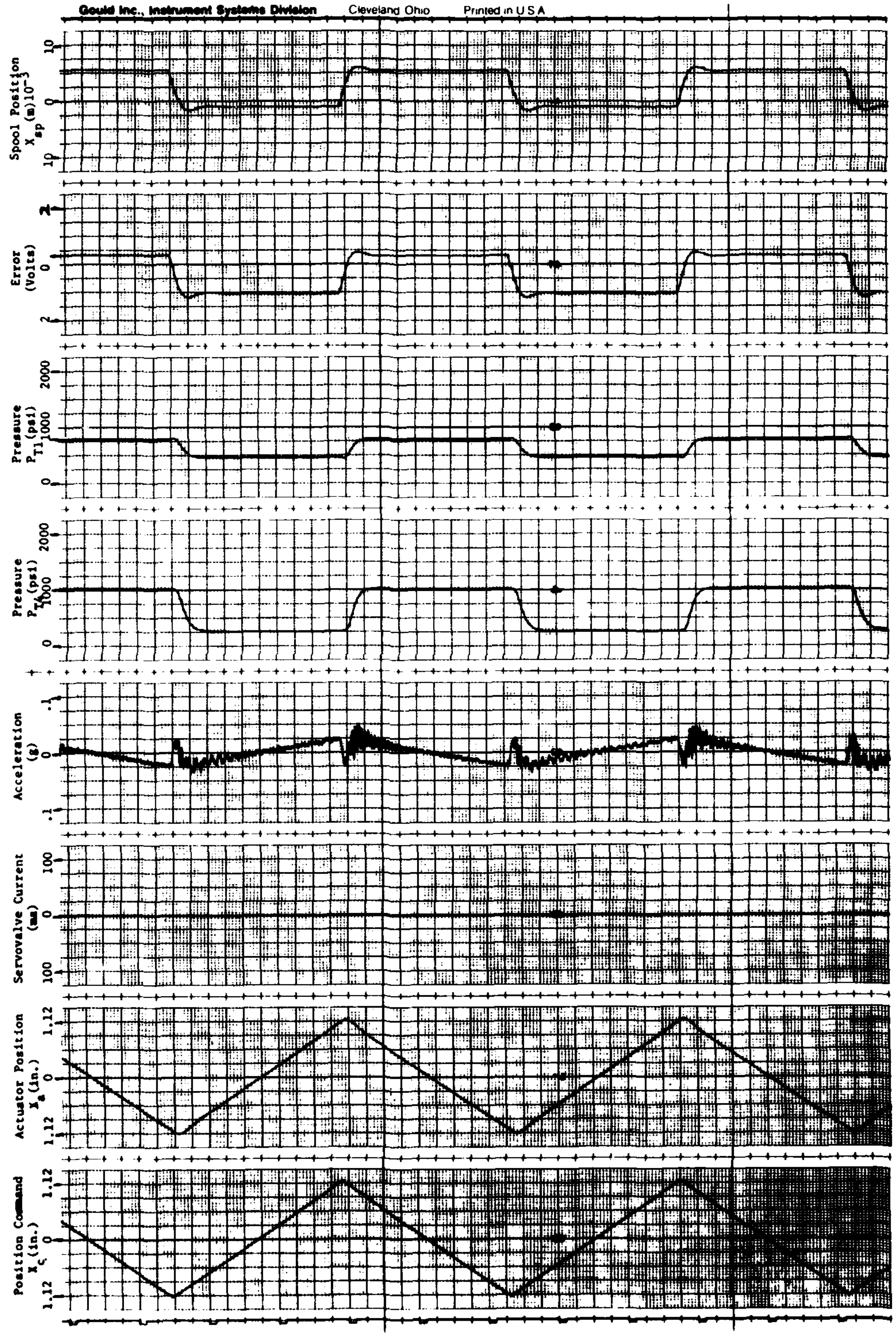

FIGURE: E-9 FREQUENCY: 0.20 WAVEFORM: TRIANGLE

VALVE GAIN: HIGH

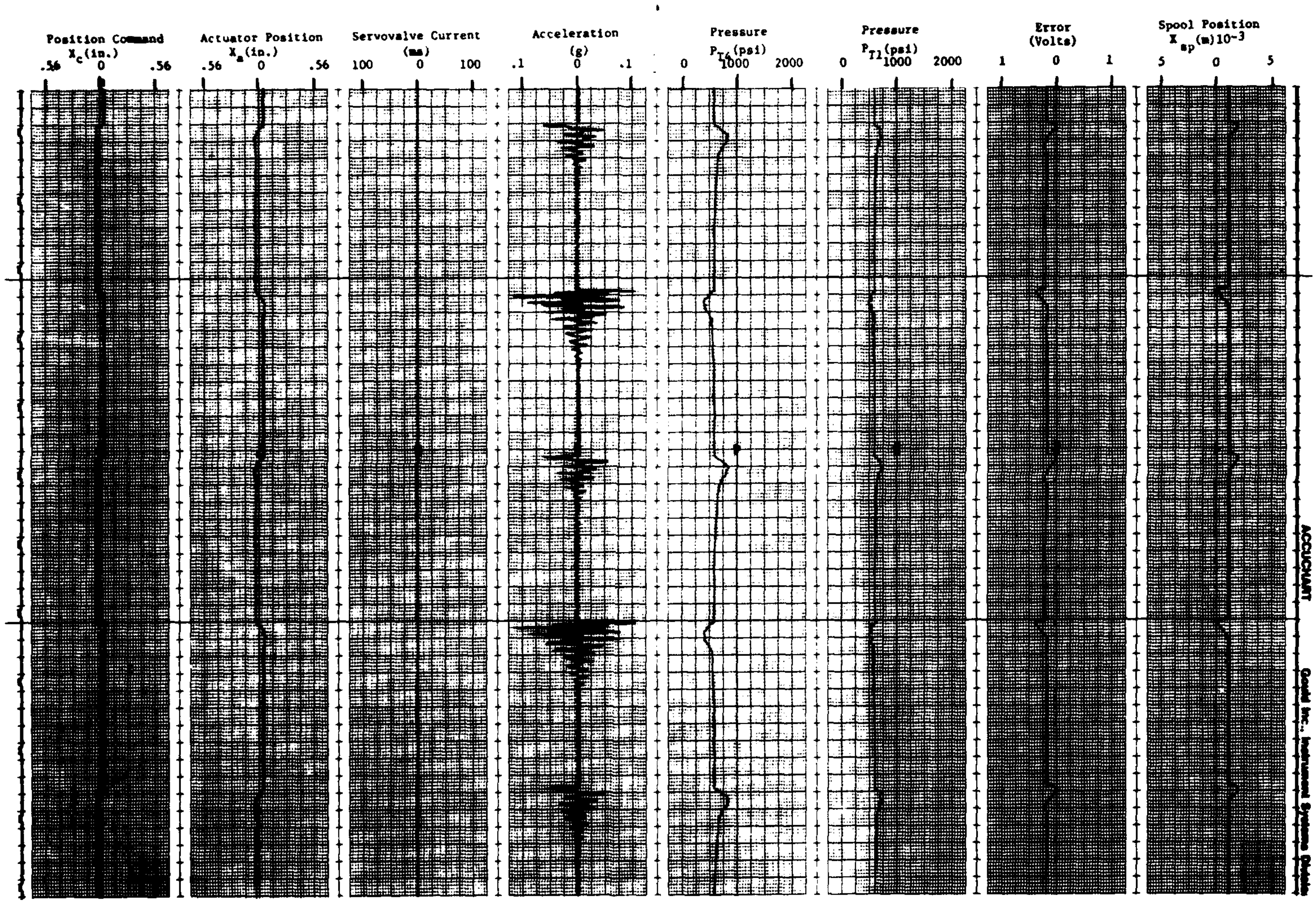

FIGURE: E-10 FREQUENCY: 0.20 WAVEFORM: SQUARE

VALVE GAIN: HIGH

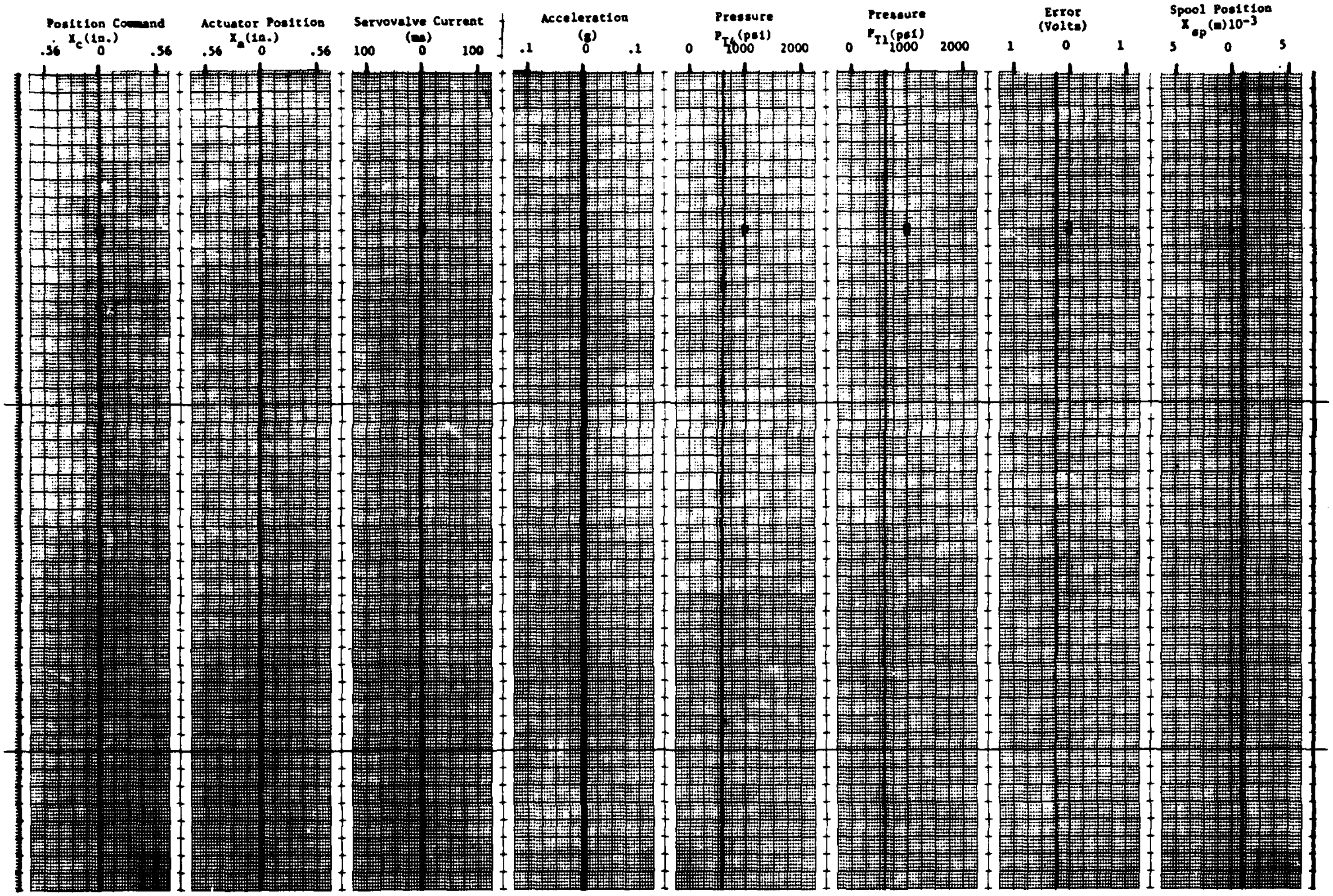

FIGURE: E-11 FREQUENCY: 0.00 WAVEFORM: ZERO

VALVE GAIN: HIGH

APPENDIX

F

Full Scale System Tests,
Franklin Low Gain Valve

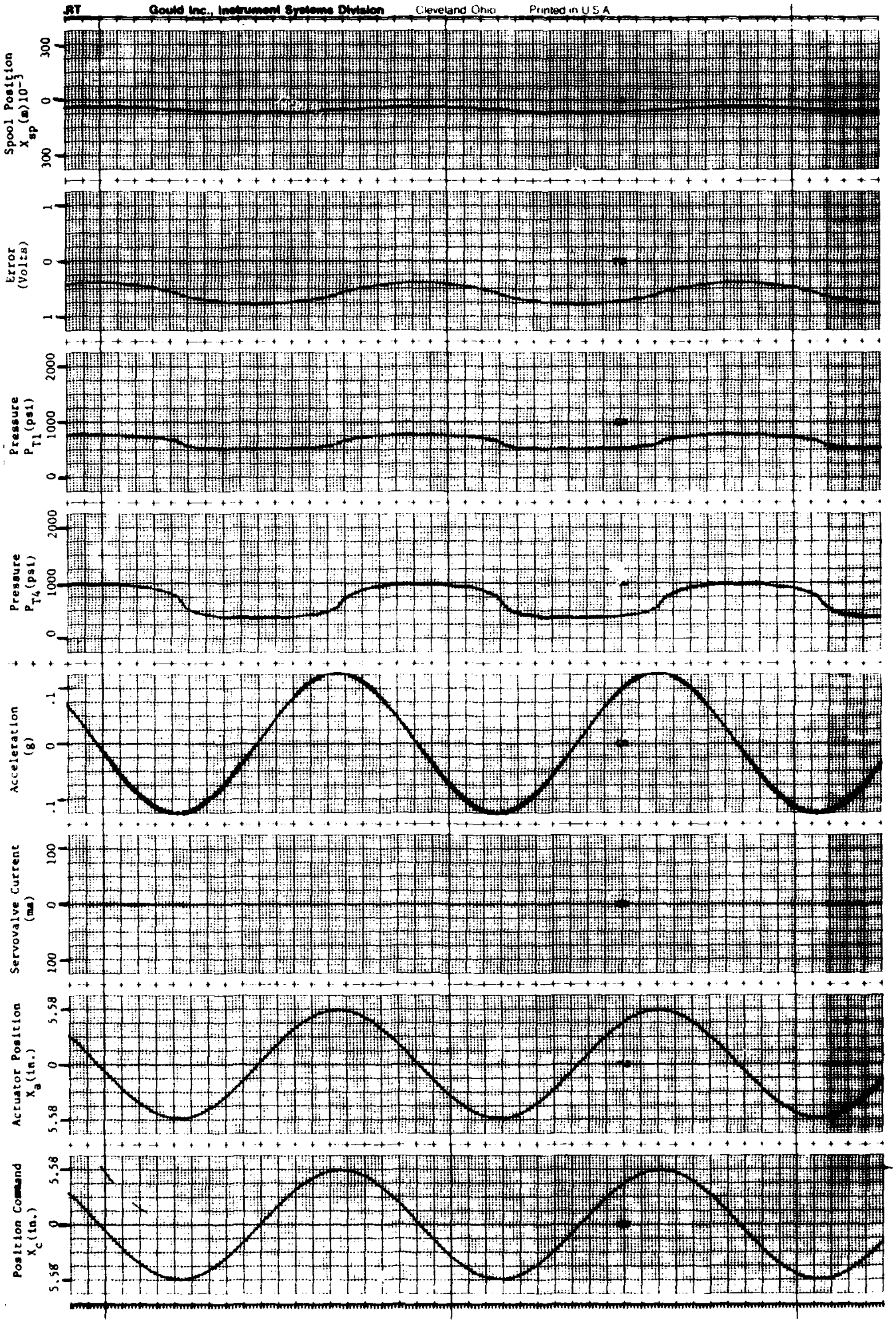

FIGURE: F-1 FREQUENCY: 0.01 WAVEFORM: SINE

VALVE GAIN: LOW

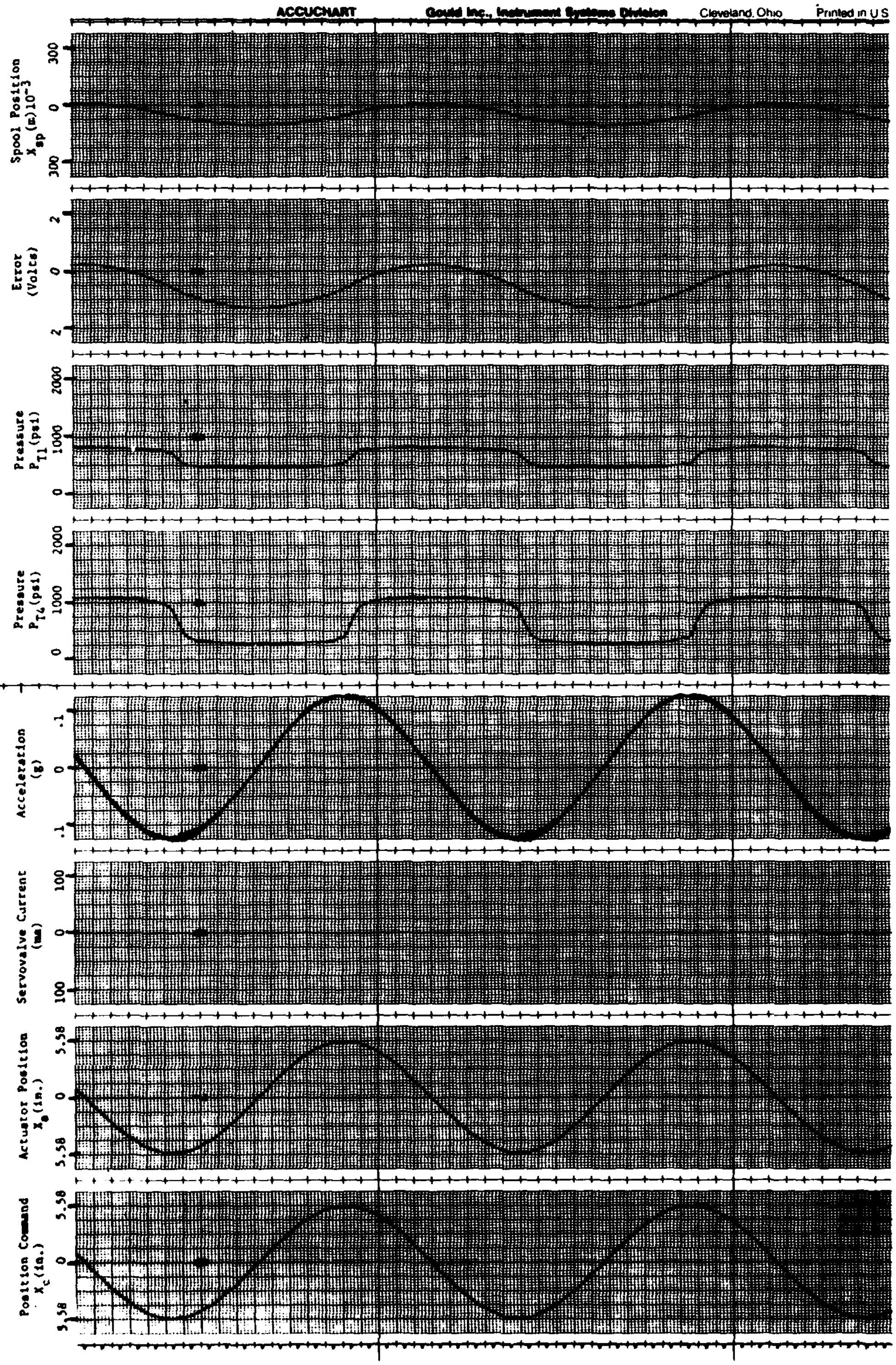

FIGURE: F-2 FREQUENCY: 0.05 WAVEFORM: SINE

VALVE GAIN: LOW

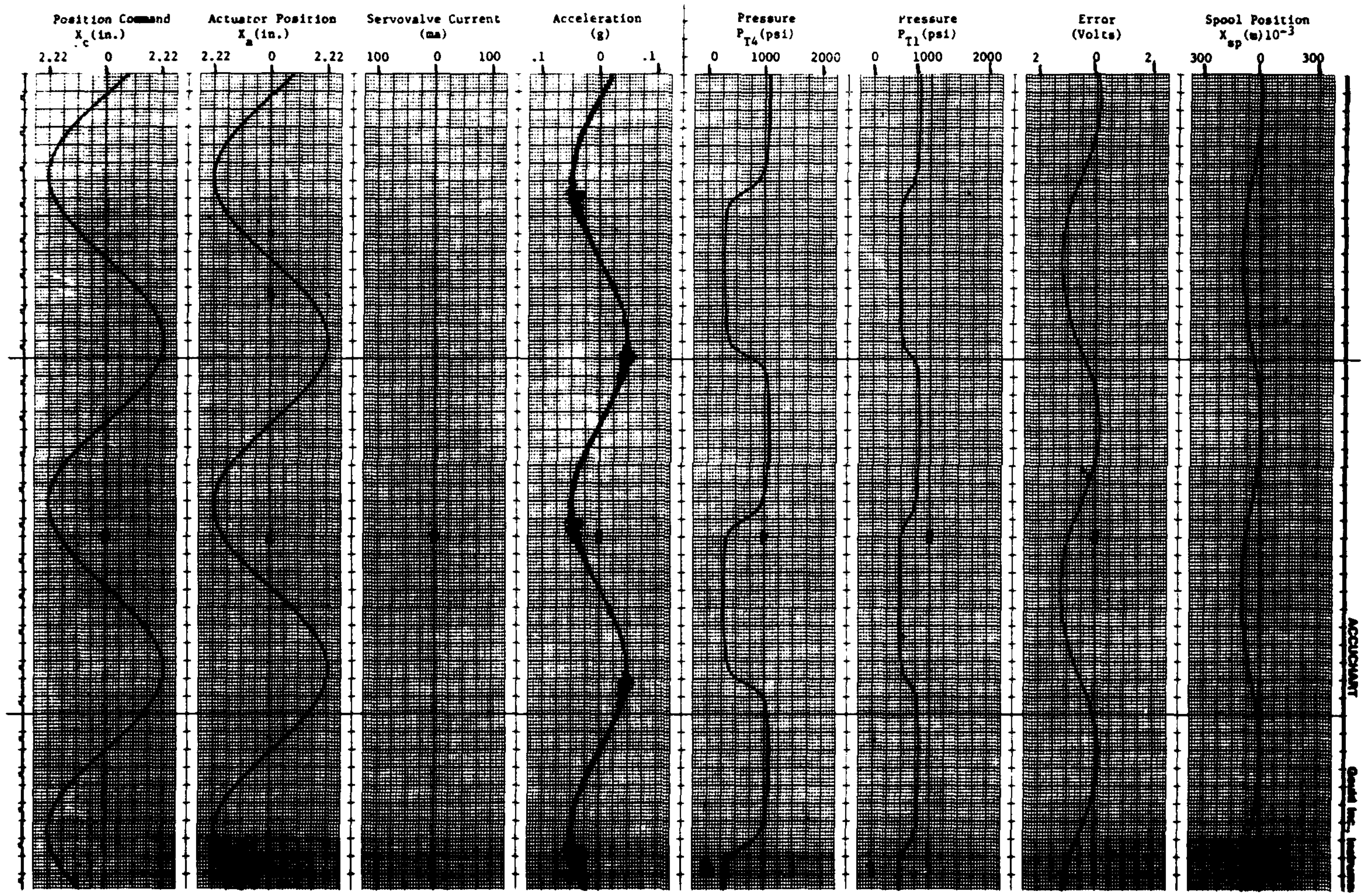

FIGURE: F-3 FREQUENCY: 0.10 WAVEFORM: SINE

VALVE GAIN: LOW

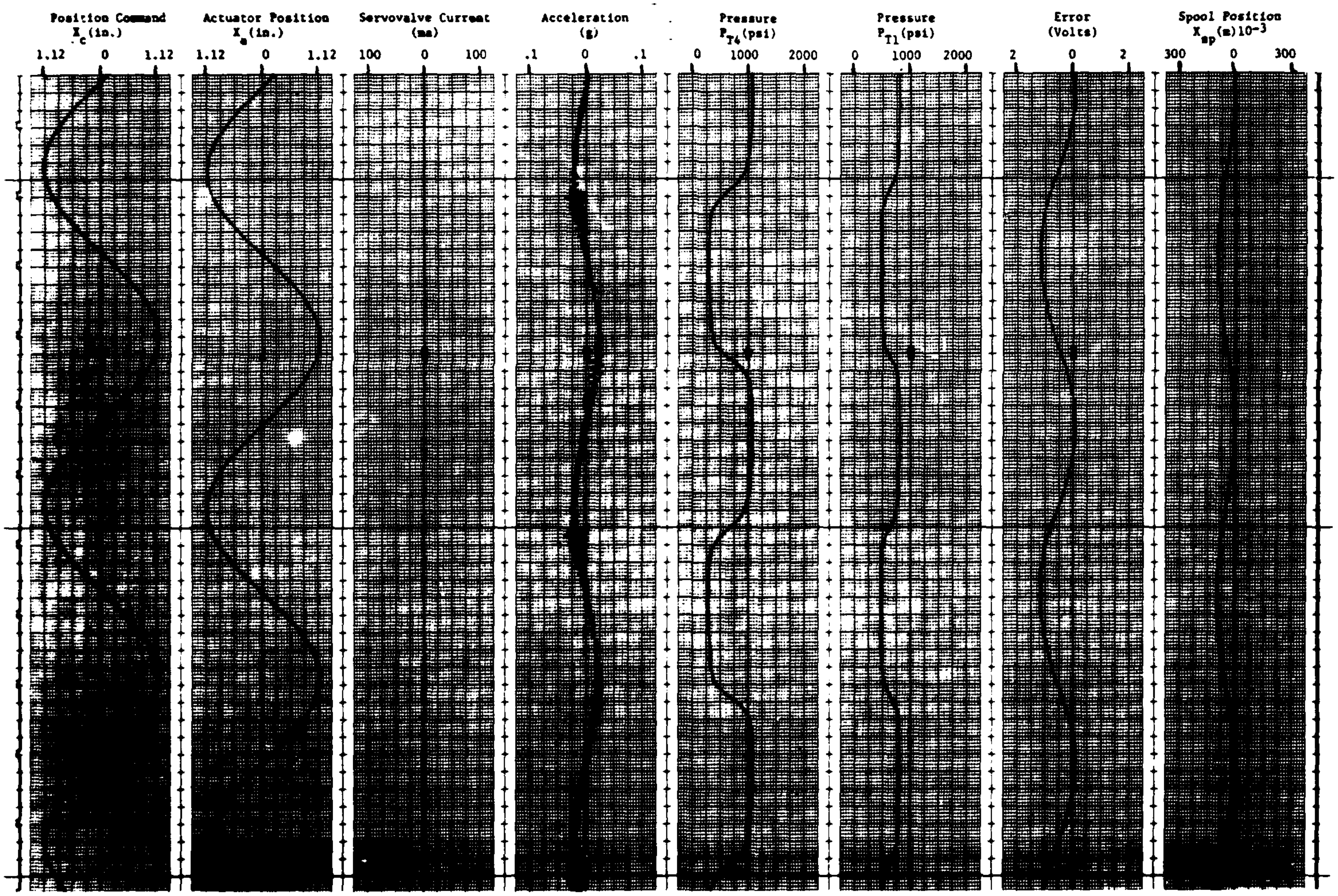

FIGURE: F-4 FREQUENCY: 0.20 WAVEFORM: SINE

VALVE GAIN: LOW

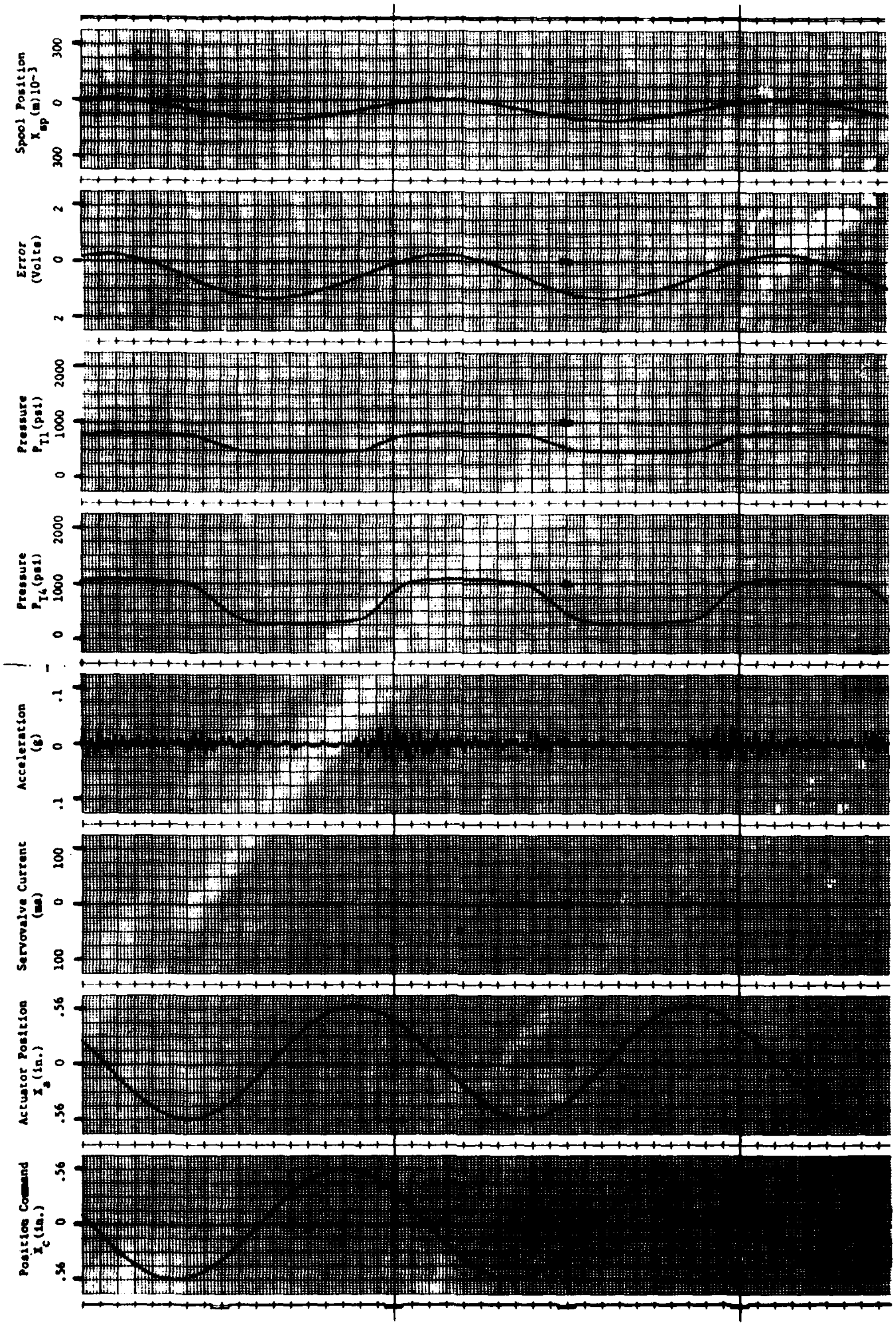

FIGURE: F-5 FREQUENCY: 0.50 WAVEFORM: SINE

VALVE GAIN: LOW

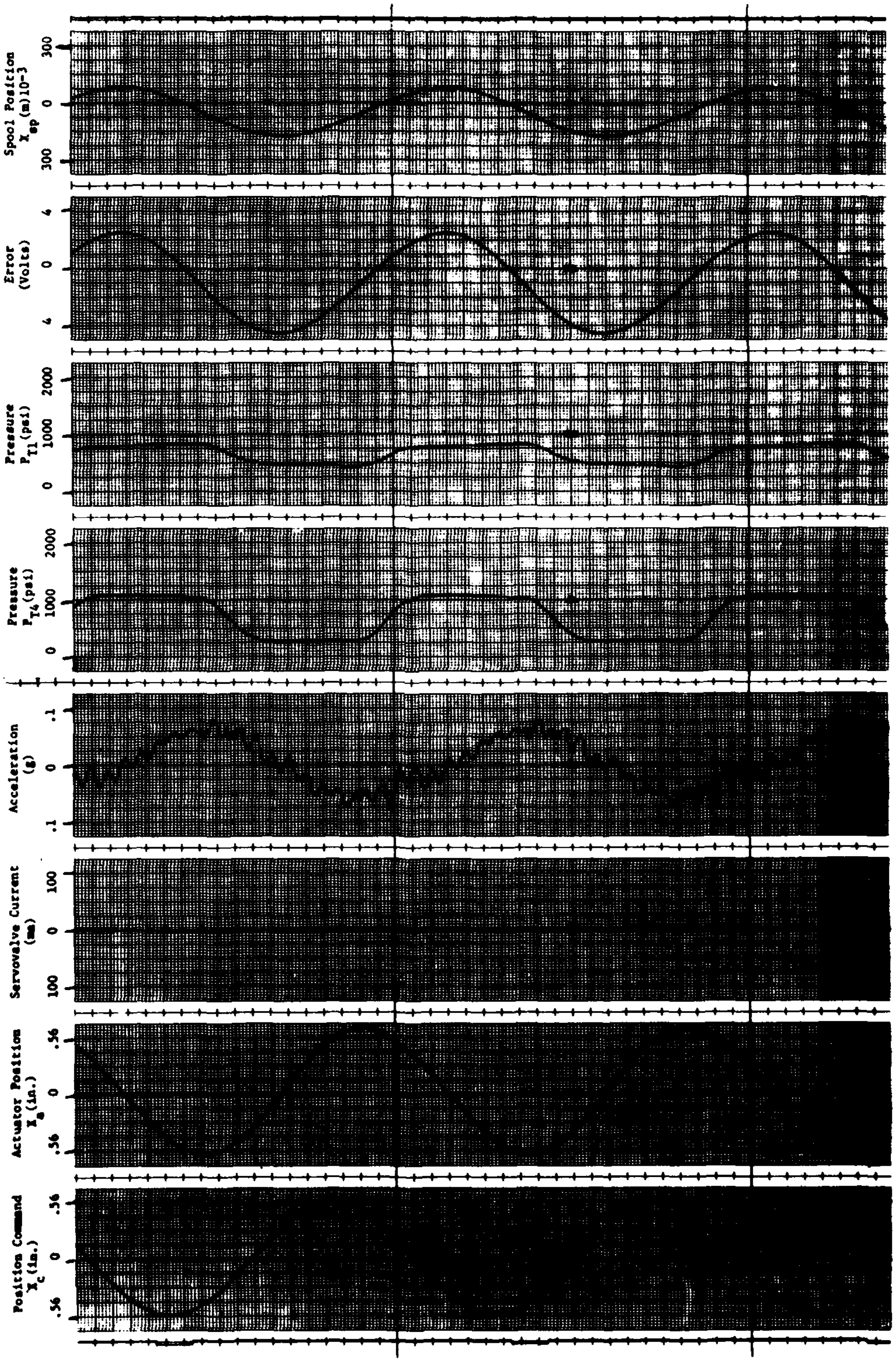

FIGURE: F-6 FREQUENCY: 1.0 WAVEFORM: SINE

VALVE GAIN: LOW

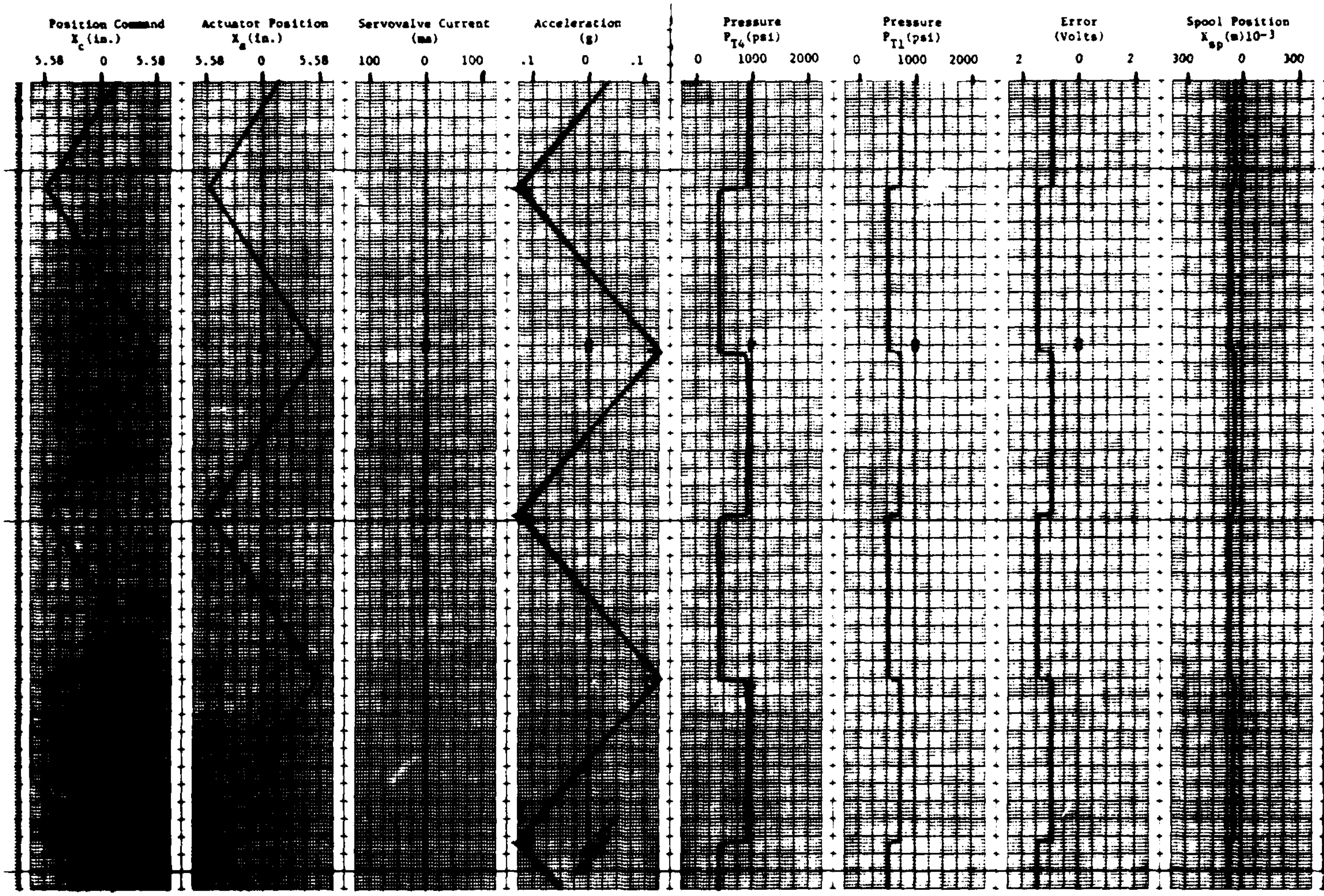

FIGURE: F-7 FREQUENCY: 0.01 WAVEFORM: TRIANGLE

VALVE GAIN: LOW

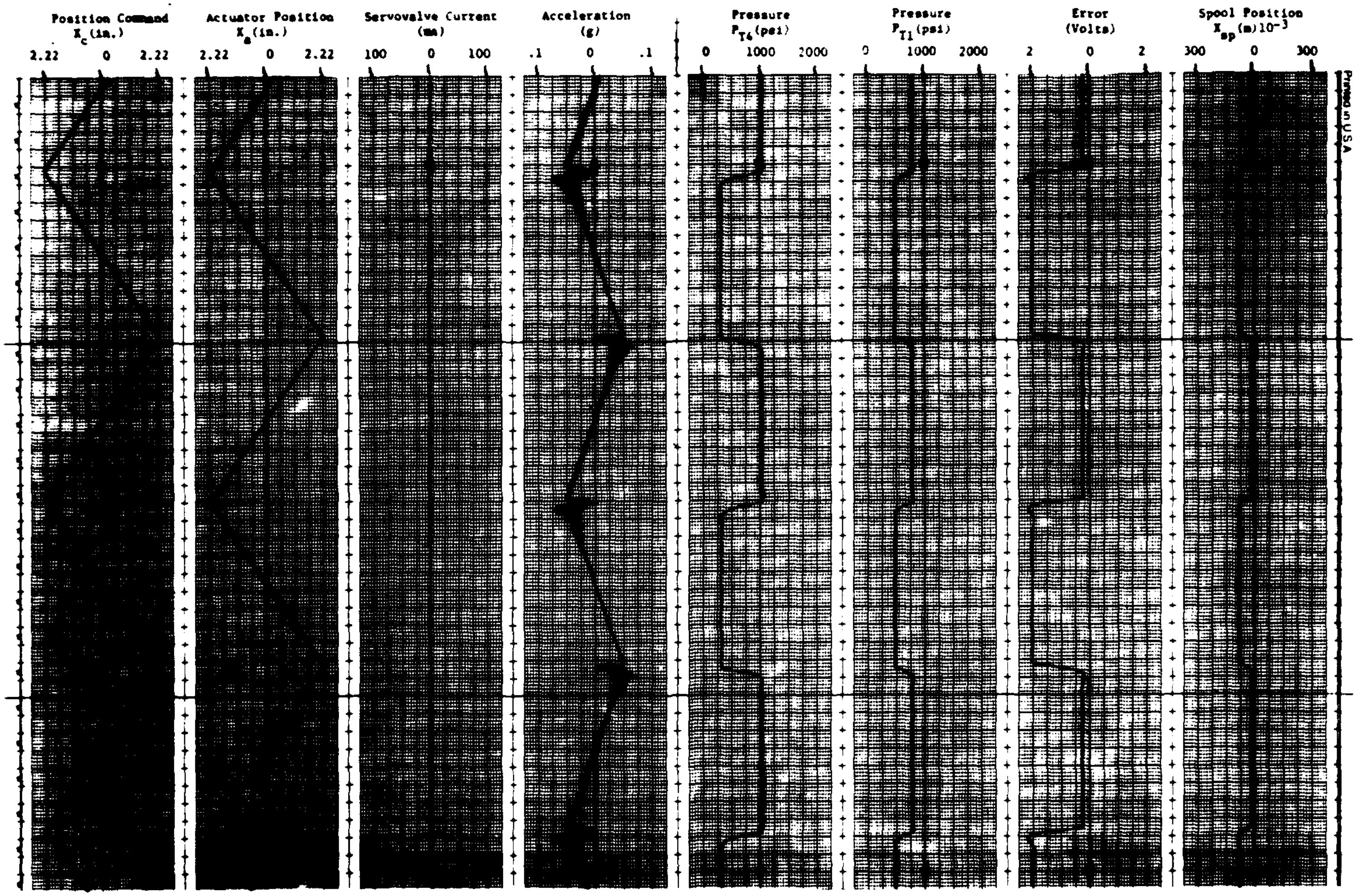

FIGURE: F-8 FREQUENCY: 0.10 WAVEFORM: TRIANGULAR

VALVE GAIN: LOW

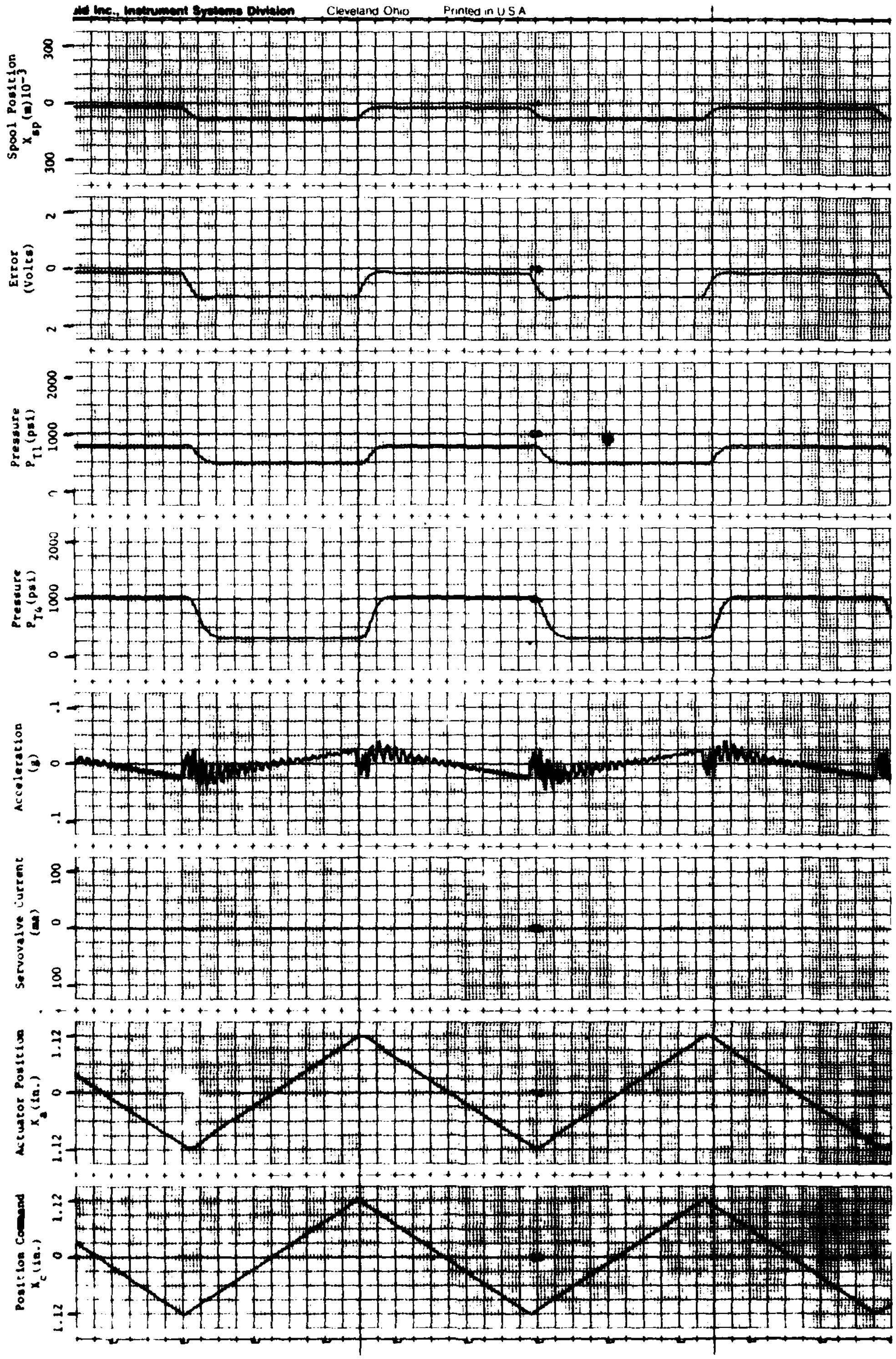

FIGURE: F-9 FREQUENCY: 0.20 WAVEFORM: TRIANGULAR

VALVE GAIN: LOW

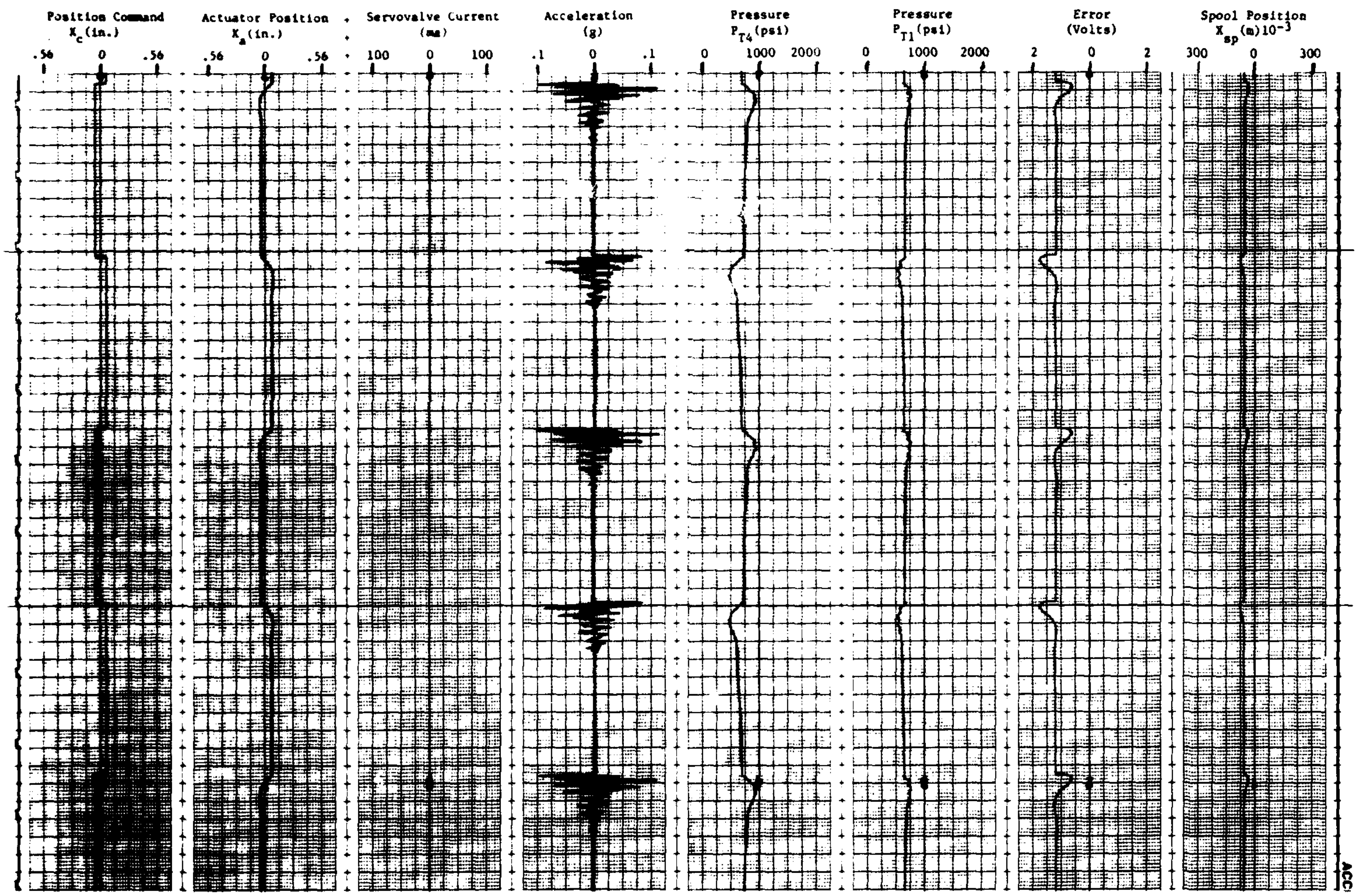

FIGURE: F-10 FREQUENCY: 0.20 WAVEFORM: SQUARE

VALVE GAIN: LOW

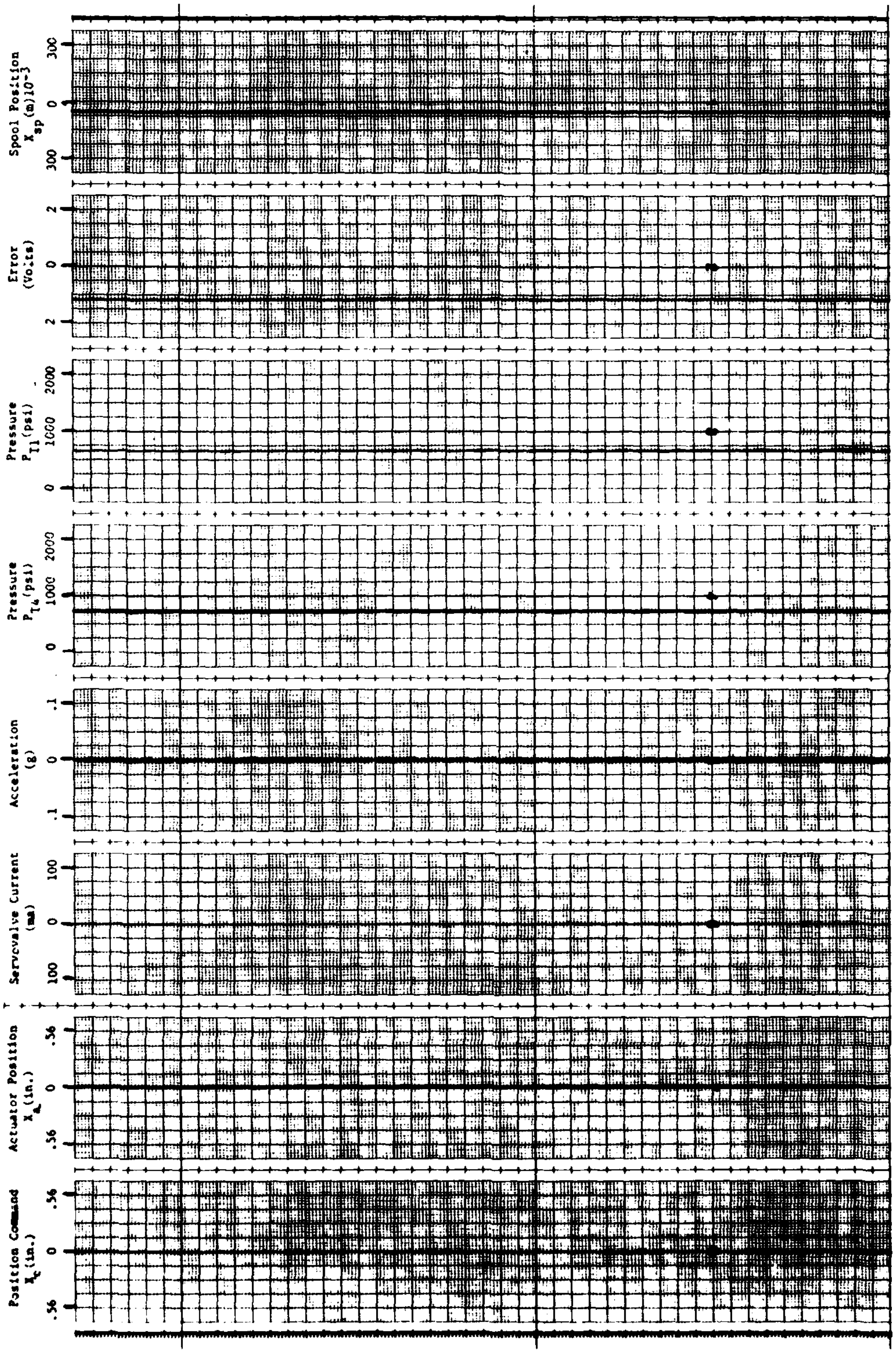

FIGURE: F-11 FREQUENCY: 0.00 WAVEFORM: ZERO

VALVE GAIN: LOW

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